POWER

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"Push, Don't Knock"

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Newark, N.J.

A FAMILIAR PHRASE *that has many times stared at us from a conspicuous position upon a door; three simple words that are almost invariably passed by with entire thoughtlessness, attributing the significance directly to only the entrance thus slightly guarded.*

BUT stop to analyze, consider more fully the meaning beyond the external marking and we find that the expression strikingly pertains to each one of us and our daily endeavors, conveying a teaching of sterling worth.

"Push, Don't Knock" has its limits on the door of the office, it has *no* limits on the door of our future. It is erasible from the former, but it is stamped *indelibly* on the latter.

Push at the door of the office and it opens just so far; push at the door of personal advancement, and the more persistent and energetic the push, the wider does it extend.

In this sense, *push* is *progress*, and progress is *going forward*. The engineer cannot remain as stationary as his engines, he *must* move, either slipping *up* or *down* the ladder of bigger things.

There are two general classes of men, the *pushers* and the *pushless*, one drives forward and keeps his step, the other *is driven* forward and falls backward. One *uses* his best efforts, the other *abuses* them.

The motto of the progressive engineer commences with "Push" and includes "Don't Knock." This man doesn't cry for a chance for advancement, he tries for it, and with that real ambition and confidence in his own ability that gets there. The engineer with push in his makeup embraces every opportunity offered him, and if it *isn't* offered, he *makes* it, he shows right in his own plant what *he* can do.

Opportunity is like the live steam line under heavy load, it's never empty, it's never used up. The pressure back of it is the enthusiasm that is put into the work of every day. That is the real power on the job.

To *knock* is the easiest thing on earth; everyone can grumble, whine and complain and "knock" his associates, his boss and his plant. It doesn't require any real ability to do this—it's boy's play.

But did you ever see an engineer who is alive to every chance for advancement spend valuable time in "knocking"? He's too busy with something to waste his efforts in nothing.

The long running hit in the game of life is the real demonstration of your real superiority. Consider it from any angle, you will find that it is this alone that counts.

To "Push, Don't Knock" is the creed of energy; it's the small beginning that offers a big ending; it doesn't require deep thinking to see it. It's the handwriting on the door of opportunity for the man who will push it open wider and wider without knocking.

Federal Power House, Boise, Idaho

By A. P. Connor

SYNOPSIS—This hydro-electric power plant has been built to supply electrical energy for constructing the Arrow Rock dam about fifteen miles distant. There are three 625-kv.-a. vertical-type generators, each with its exciter coupled to the upper end of the generator shaft. The voltage of the main units is 2300, which is stepped up to 22,000 volts for transmission. When the dam is completed the plant will be used for supplying energy for local use.

The project which the United States Reclamation Service is undertaking for irrigation purposes at Boise, Idaho, has in reality three natural subdivisions, one being in the valley of the Payette, the second in the Boise Valley and the third in the valley on the north side of the Boise River. The reference in this article is to the part of the project which makes use of the Boise River for power purposes.

The dam across the Boise River, Fig. 1, which makes its waters available for irrigation and power purposes, is situated about eight miles above the City of Boise. The dam raises the water level 33 ft. and diverts water into a canal extending 23 miles to Indian Creek, the channel of which is then used for about nine miles, where the water is then directed into another canal, eight miles long, and which terminates in the Deer Flat Reservoir. The possible power development is estimated at 15,000 hp. The power possibilities from the various streams in the vicinity of the project are great, and a number of hydro-electric plants have been constructed.

The turbines of the Boise hydro-electric plant, Fig. 2, rest on the discharge tunnels, thus utilizing short draft tubes, Fig. 3. The thrust bearings are just above the wheel pits, and the alternators rest on the main floor of the generator room. There are three vertical-type, alternating-current generators, each of 625-kv.-a. capacity, with exciters mounted on the end of each shaft. The units generate 2300-volt three-phase current at 60 cycles, and this is transformed to 22,000 volts for transmission to the Arrow Rock dam. The generators are synchronized on the 22,000-volt side of the transformers, and provision is made for operating them in parallel. The generators are of the revolving-field type and have a rating of 500 kw. at 80 per cent. power factor with a speed of 180 r.p.m.

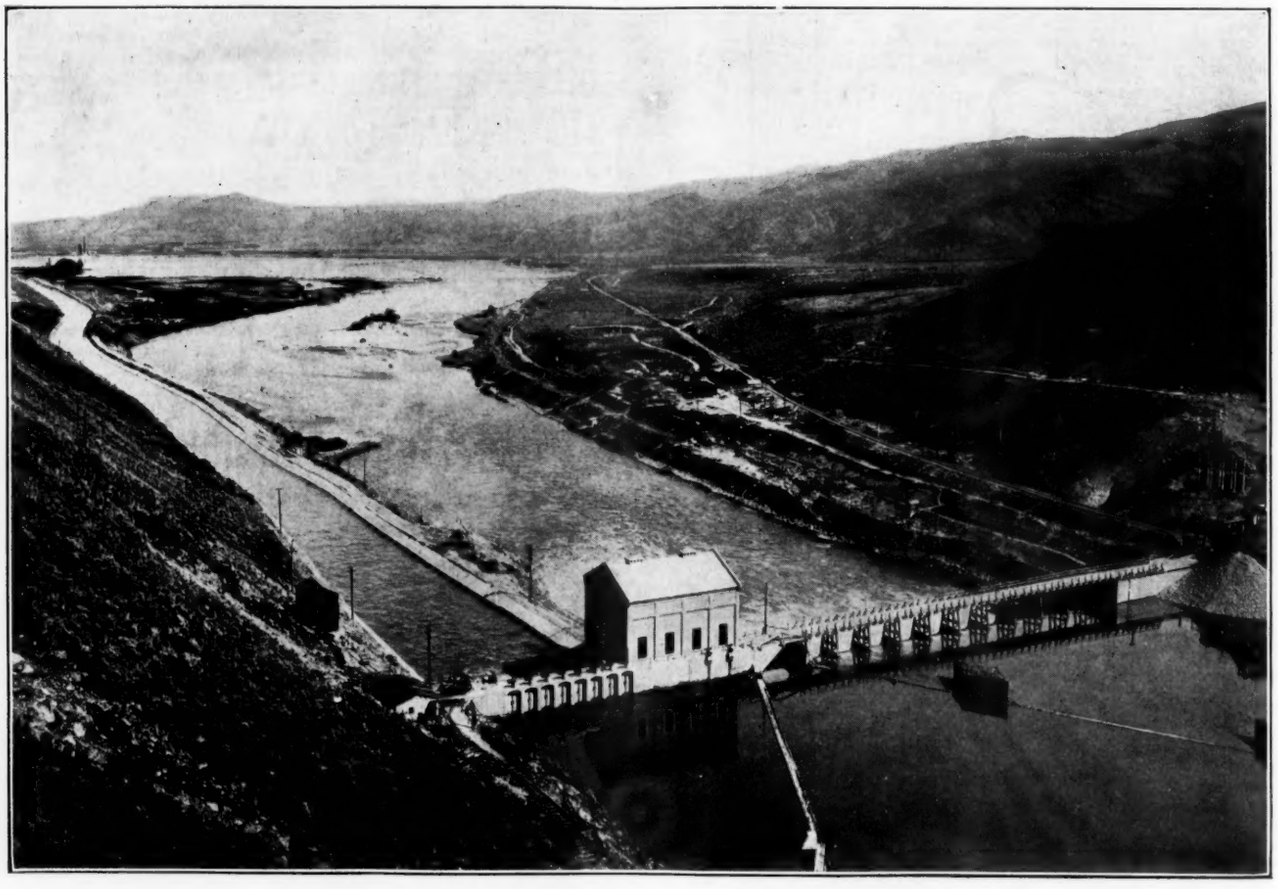


Fig. 1. Dam and Power House at Boise River

The exciters are supported by the top spiders of the respective alternators. The capacity of each is sufficient to furnish the maximum field current for two alternators, plus 10 kw. for local or other requirements. The exciters are designed to generate direct current at 125 volts. A regulator is used to control the generator voltage. Fig. 4 is a plan of the plant.

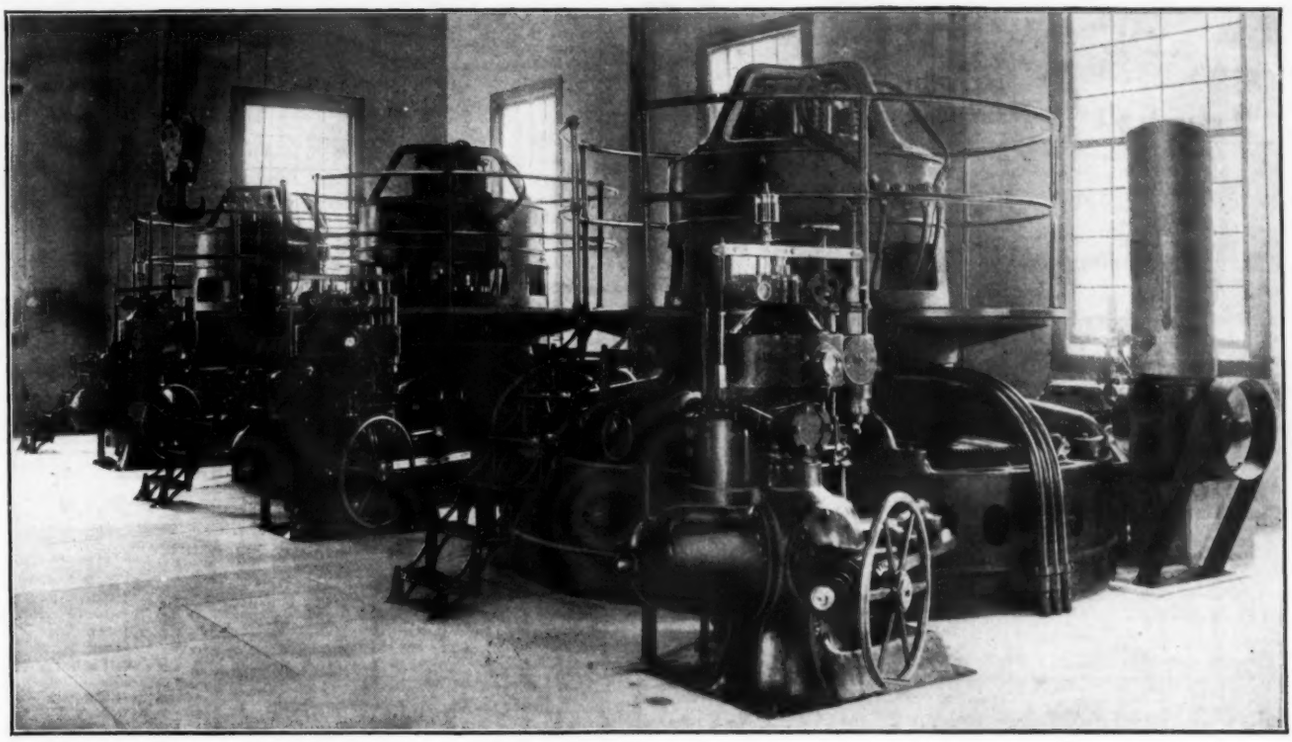


FIG. 2. GENERATING UNITS OF THE BOISE RIVER POWER PLANT

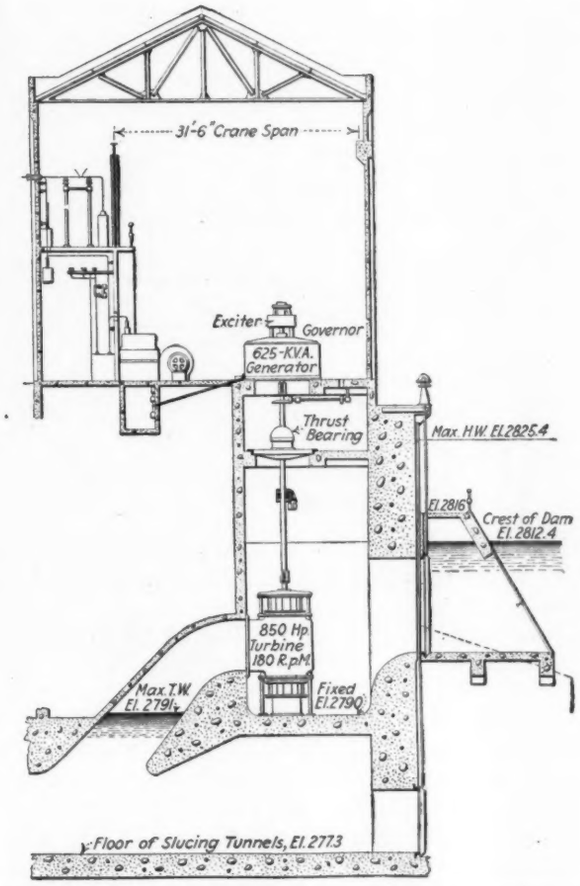


FIG. 3. ELEVATION OF ONE OF THE TURBO-GENERATOR SETS

The transformers for the alternators are air cooled by motor-driven blowers. An air compressor having a capacity of 50 cu.ft. per min. is provided for the needs of the station, and is driven by a 220-volt induction motor.

The present capacity of the plant is 1500 kw. A certain amount of electricity is used for local purposes on the dam, and some is used by the surrounding community, but the greater amount is intended and used for the building of the dam some fifteen miles distant. Nevertheless, the plant is a permanent structure and it is proposed to turn

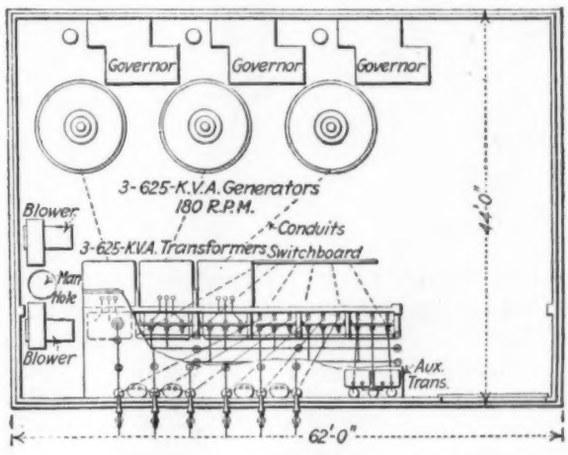


FIG. 4. PLAN OF THE BOISE POWER PLANT

the power over to local uses when the Arrow Rock dam is finished.

The plant is compact, the size of the power house being about sixty by forty feet. The irrigation canals which the dams in the project supply with water are in all about four hundred miles in aggregate length. The laterals from them are in the aggregate one thousand miles, an indication that the project is of more than ordinary consequence and magnitude.

Steam-Turbine Diagrams

By F. R. Low

SYNOPSIS—An explanation, "written so you can understand it," of the diagrams which show the working of the steam in turbine blades and how the energy of the swiftly moving jet is absorbed and converted.

To one riding in a railway car a ball traveling along parallel to the track, as in the line *mn*, Fig. 1, and with the same velocity as the car would appear to be stationary relatively to the car, just as though it were attached thereto by an invisible rod. If the car slowed up, the ball, although preserving its own velocity, would appear to shoot ahead, while if the car speeded up the ball would appear to be moving backward, with a velocity in either case equal to the difference between its own speed and that of the car.

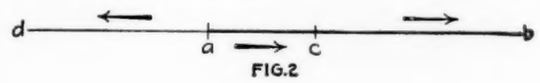


FIG. 2

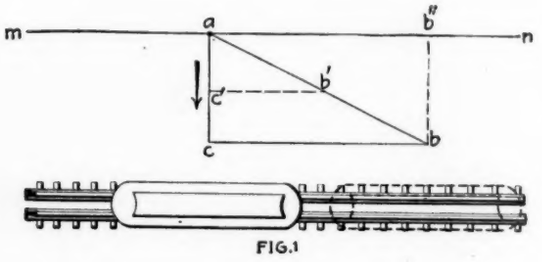


FIG. 1

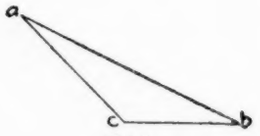


FIG. 3

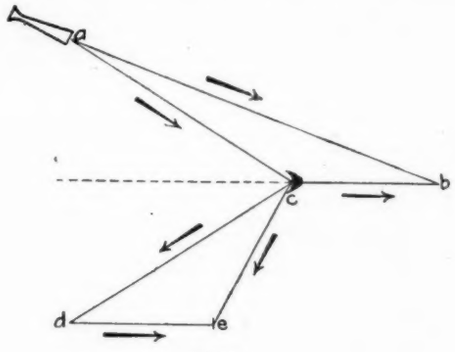


FIG. 4

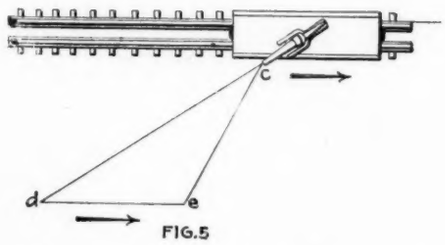


FIG. 5

Let the line *ab*, Fig. 2, represent in length and direction the velocity and direction of the ball. Set off upon it a distance *cb* representing to the same scale the velocity of the car at any instant. Then *ac*, the difference of these velocities, will be proportional to the speed of the ball relatively to the car; that is, to the velocity with which it would appear to the man in the car to be moving. Suppose the velocity of the car to be increased to that represented by the line *db*; then to the man in the car the ball would appear to be moving backward with a velocity proportional to *da*.

Returning to Fig. 1, suppose something should give the ball an impulse in the direction *ac*, with a velocity proportional to the length of that line, the velocity of the car and the previous velocity of the ball in the direction *mn* being proportional to $ab'' = cb$. To the man in the car the ball would appear to be coming straight toward him, just as though the car were standing still and the ball were batted from rest. And it would be coming straight toward him, for its motion relatively to the car would be in the line *ac*; but as the ball is going forward at the same time that it is moving sidewise toward the car, its path relatively to the ground would be *ab*, just as though it had gone to *b″* first with its original velocity and direction, and then from *b″* to *b*; but the car has gone ahead at the same time, so that it would hit it in the same spot and with the same velocity and at the same right angle as though both car and ball were standing still when the ball was impelled in the direction *ac*. At the end of the first half-second the ball, if moving only in the direction *ac*, would have gone from *a* to *c′*, one-half of *a* to *c*, but in the same time, on account of its movement in the direction parallel to the track, would have gone from *c′* to *b′*, so that at the end of the half-second it would be found at *b′*; and so for any subdivision of the time it would be found upon the line *ab*, which is the path that it would follow relatively to the ground.

If we represent, then, by the line *ab* the velocity and direction of the ball relatively to the ground, and by the line *cb* the direction and velocity of the car, the line *ac* joining their extremities will represent in length and direction the velocity and direction with and in which the ball approaches the car and with and from which it would appear to the man in the car to be coming.

With the ball still moving in the line *ab* and with a velocity proportional to the length of that line, suppose the velocity of the car to be reduced to *cb*, Fig. 3. Then the ball, although moving in the same direction and with the same speed relatively to the ground as before, will appear to the man in the car to be coming at him in the direction *ac*, Fig. 3, and with a velocity proportional to the length of that line.

Applying this to the relative motion of steam in a turbine, let *ab*, Fig. 4, represent, by its direction and length, the direction and velocity of a jet of steam issuing from a nozzle. Let *cb* represent the direction and velocity in and with which the blade moves. Then the jet would approach the blade in the direction *ac* and impinge upon it with a velocity proportional to the length of that line.

If the blade were symmetrical, so as to turn the jet back at the same angle, the jet would leave the blade in the direction *cd* and—neglecting loss from friction, impact, etc.—with the same velocity, $cd = ac$.

Suppose a ball to be fired from a gun placed at an angle upon a car, as in Fig. 5. If the car were standing still

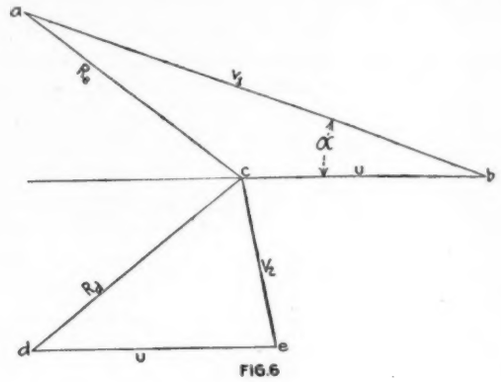


FIG.6

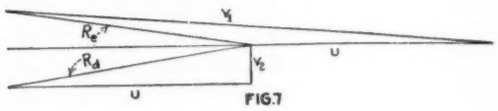


FIG.7

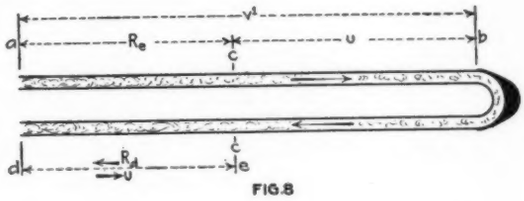


FIG.8

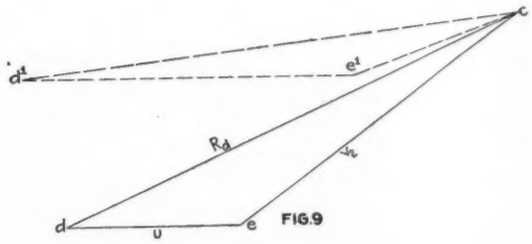


FIG.9

the ball would go off in the line *cd* relatively to the ground, but if the car were moving with a velocity which was to that with which the ball is projected as *de* is to *cd*, the ball would, by reason of the velocity acquired as a part of the car's contents, move forward a distance proportional to *de* in the same time that it moved the distance *cd* in the direction of its projection. Instead of being at *d*, therefore, it would be at *e*, and the path that it would have followed relatively to the ground would be *ce*.

In Fig. 4 we have a similar case. The steam is coming out from the blade in the direction and with the velocity *cd*, but it is traveling with the blade with the velocity and in the direction $cb = de$. Set off *de*, equal in length and parallel to *cb*, and the line *ce* will represent the direction and velocity of the steam when it leaves the blade with respect to the ground or to the stationary nozzle.

Fig. 4 or 6 is the typical diagram for a single-stage impulse turbine. The initial velocity is V_1 and the final velocity, V_2. The smaller V_2 in relation to V_1, the greater the proportion of the energy of the jet which has been absorbed by the turbine.

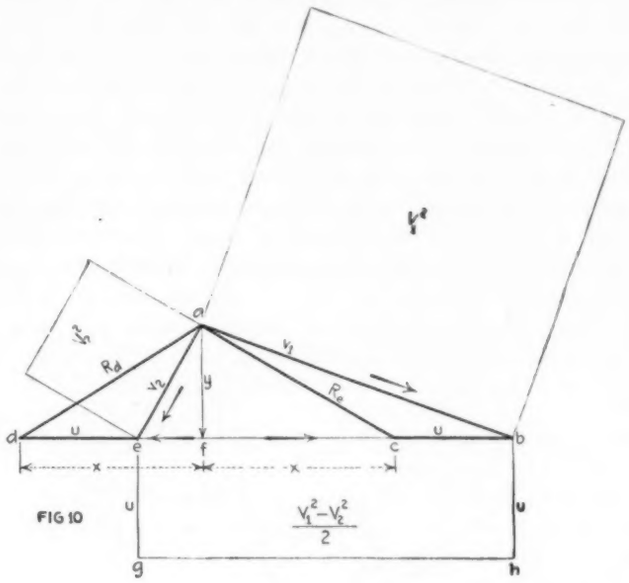


FIG 10

So long as the jet approaches the blade at an angle, as at *a*, there will be some sidewise direction to the final velocity V_2. As the angle *a* becomes less, the smaller may V_2 become, as in Fig. 7, until the jet is in line with the blade, Fig. 8, when the sidewise component disappears altogether and the diagram becomes a straight line, as shown in that figure. Its full length, $ab = V_1$, represents

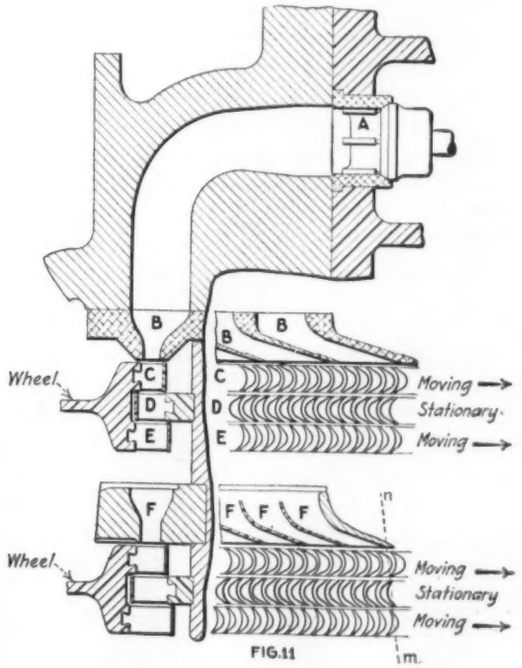


FIG.11

the initial velocity; $cb = u$, the blade velocity, and $ac = R_e$, the relative velocity and direction of entry, just as it did in the preceding figure, only that the triangle has been closed up and *c* lies upon the line *ab*.

If the jet is going at, say 1000 ft. per sec., and the blade is running away from it at 500 ft. per sec., the

jet will be overtaking the blade at the rate (that is approaching it with the relative velocity) of 1000 — 500, or $R_e = V_1 - u = ab - cb = 500$ ft. per sec. This is R_e, the relative velocity with which the jet enters the blade. Neglecting friction, the jet is reversed in the blade and thrown backward to the same velocity R_d relatively to the blade; but, like the ball that was fired backward from the car in Fig. 4, it must have its forward motion subtracted from its backward velocity to show its velocity relatively to the ground. And when, from the relative velocity of discharge R_d, Fig. 8, we subtract the blade velocity u, by setting off upon the line $dc = R_d$ the line $de = u$, we have nothing left, indicating that velocity and energy have all been abstracted, that the steam has been brought completely to rest.

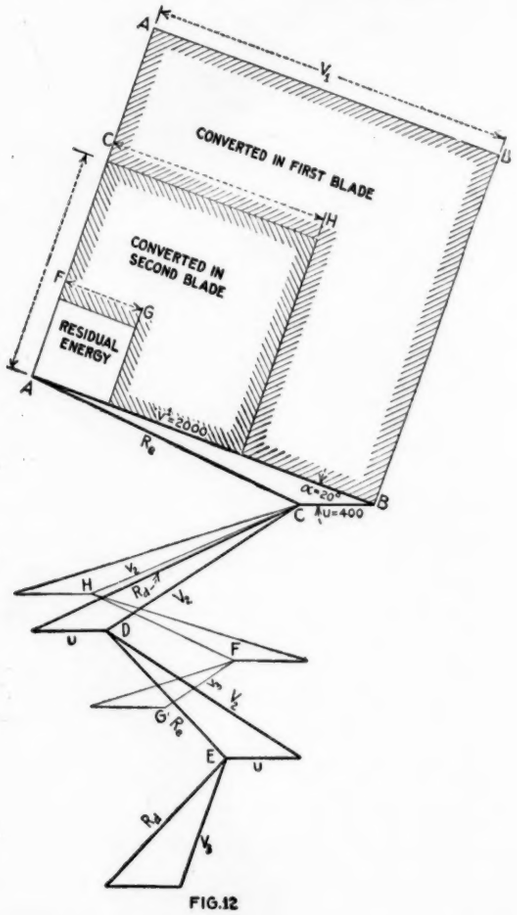


FIG. 12

In order to do this for the abstract case the blade velocity u must be one-half the initial absolute velocity V_1, Fig. 7; for in order that the blade velocity $u = de$, Fig. 8, may equal and cancel the relative discharge velocity $R_d = cd$, u must also equal R_e, to which R_d is also equal; and since ab equals $R_e + u$, each of these quantities, if they are equal, must be one-half of ab.

Fig. 9 is the typical diagram for a single stage of a reaction turbine; $cd = R_d$ represents the direction and velocity with which the steam is discharged from the blade; u is the blade velocity, and V_2 the velocity relatively to the ground, as in the other diagram. So long as the jet is delivered at an angle there will be some residual velocity V_2 in the jet. By reducing the angle and increasing the blade speed, as shown in the dotted position, this residual velocity could be reduced from ec to $e'c$, but it is only when de becomes parallel with and equal in length to dc that the residual velocity disappears altogether. The blade of a reaction turbine must run at the same velocity as the jet, in order to reduce its velocity to zero and extract all of its energy. If a car were paying out a cable slower than the car was running, it would be dragging the cable along the ground. If it were paying the cable out faster than the car was running, it would be pushing the cable backward. If the speed of the cable through the opening in the car equaled the forward movement of the car, the cable would be laid quietly upon the ground with all the velocity and momentum which it possessed, as a part of the car's contents, taken out of it.

In Fig. 10 the triangle $cdec$ has been moved from the position which it occupies in Figs. 4 and 6 and attached to the other triangle at a instead of at c, producing the diagram which, as shown in the firmer lines, is frequently seen in turbine literature. The jet travels a distance proportional to ab in a second, in the direction in which it is discharged, but moves only a distance proportional to fb in the direction db in which the blade is moving.

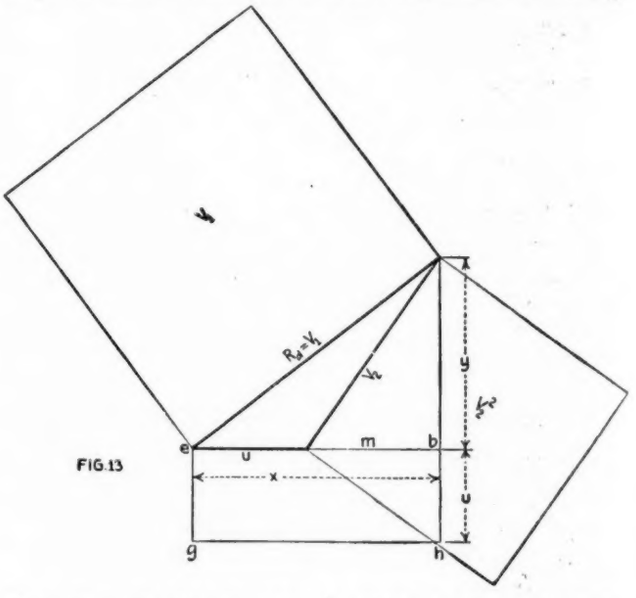


FIG. 13

This line fb (the length of which is determined by dropping upon it a perpendicular as af from the other extremity of the line to one extremity of which it is connected) represents the "component" in the direction db of the velocity represented by ab. This component of the entering velocity is fb; the component of the discharge velocity ae is ef, opposite in direction from fb. The blade, therefore, has overcome, or annulled, the velocity fb and has gotten up new velocity fe. In other words, it has retarded the steam enough to destroy the velocity fb and accelerated it enough to produce the velocity fe. But retardation is only negative acceleration, and the effect is the same as though the steam had been accelerated from rest to a velocity proportional to be.

The *force* which must be exerted upon a body to produce a change in the velocity of its motion is equal to the product of its mass and the acceleration produced.

The *energy* is equal to the product of the force and the space through which it is exerted.

The energy produced by the flowing through the blades of a given mass of steam per second will be

$$\text{Energy} = \underbrace{\text{mass} \times \text{acceleration}}_{\text{(force)}} \times \underbrace{\text{blade speed}}_{\text{(space)}}$$

The acceleration is *be* and the blade speed *u*, equal to *bh* or *eg*, so that the area of the rectangle *ebhge* equals the product of the acceleration and blade speed, and the energy absorbed by the blade equals *mass* × area *ebhge;* that is, is directly proportional to that area.

The energy stored in a moving body is the product of the mass and one-half the square of its velocity. The energy in the entering steam is proportional to *mass* $\times \frac{V_1^2}{2}$, or to the area of the square upon the line *ab* multiplied by one-half of the mass. The residual energy in the escaping steam is *mass* $\times \frac{V_2^2}{2}$, or the area of the square erected upon the line *ae* multiplied by one-half the mass. The difference, *mass* $\times \frac{V_1^2 - V_2^2}{2}$, represents the difference in the energy of the steam as it enters and leaves the blade. But, neglecting friction, etc., there is no other place for the energy to go than to be absorbed by the blade, and the difference ought to be equal to the energy which we found to be so absorbed; hence

$$Mass \times \frac{V_1^2 - V_2^2}{2} = mass \times ebhge$$

Or, since the mass is common to both,

$$ebhge = \frac{V_1^2 - V_2^2}{2}$$

That is, the area *ebhge* equals one-half of the difference between the areas of two squares on *ab* and *ae*, or the difference between the areas of the two squares is twice the area *ebhge*.* Notice also in passing, that the work done is directly proportional to the line *eb;* that is, to the force exerted in the direction of the blade movement.

The velocity-stage turbine abstracts a portion of the residual energy V_2 by passing the steam again through the same or another set of blades. Suppose the size and speed of the turbine were such that a blade velocity *u* of 400 ft. per sec. could be obtained. Suppose, further, that the conditions as to initial pressure, superheat and vacuum were such that the steam would attain a velocity of 4000 ft. per sec. if allowed to complete the expansion in a single stage.

If the turbine were designed so that the steam would acquire half this velocity in the first stage, there would have to be three more such stages to complete the expansion; that is, if the work were to be equally divided between the stages, or if in no stage was the velocity to exceed 2000 ft. per sec., it would have to be a 4-stage turbine. The energy in a moving body varies as the *square* of the velocity. There is only one-quarter as much energy in steam flowing at 2000 ft. per sec. as at twice that speed, so that three similar stages with velocities of 2000 ft. would have to be used to take out the other three-quarters.

The velocity decreases inversely as the square root of the number of stages—½ the velocity for 4 stages, ⅓ the velocity for 9 stages, etc.

Suppose, then, the steam is expanded in the first stage through a range that will give it a velocity of 2000 ft. per sec. The blade speed is 400 ft. per sec. and the angle *a* of the nozzle 20 deg., as shown in Fig. 11. It will be seen that there is considerable residual velocity V_2. The only way to reduce this would be to reduce the initial velocity V_1, which, as just shown, would require a number of stages varying as the square of the number of times the velocity is reduced, or to increase the blade speed *u*, which by the conditions of the case is impossible.

The somewhat common impression that reducing the blade speed and employing more stages produces better economy is not borne out by this analysis. For the abstract case the hydraulic efficiency depends upon the ratio of the steam speed to the blade speed and would be the same for one-half the blade speed if the steam speed were also halved. The turbine built with the greater number of stages would, however, be the more efficient because of the reduced surface friction, and because of the greater area of blade passage due to the lower velocity. The nozzles would occupy a greater number of degrees of the wheel surface and the gain from reheating would be slightly greater.

Fig. 11 shows diagrammatically a Curtis turbine. The high-pressure steam enters at *A* and expands in the nozzles *BBB*, impacting upon the moving blades *CC*, which are carried upon the crown of the running wheel, as shown at the left. Carried upon the casing is a set of stationary blades *DD* into which the steam is discharged with the velocity and in the direction R_d (of the diagrams), and in which the steam is turned around and discharged upon the blades *E*, also attached to the moving wheel. The blades *E* are deeper than the first ones *C*, not because the steam is supposed to expand in going through them, but to allow the constant volume of steam to pass at a decreased velocity. It comes off from the last row of blades *E* at the same pressure as that with which it left the nozzle *B*, but with its velocity very much reduced, and passes to the second set of nozzles *FFF* to have its velocity accelerated by another expansion.

If the blades were symmetrical the diagram for the abstract case would be that shown by the heavy lines in Fig. 12. The velocity V_2 with which the steam leaves the blade *C* is the same with which it enters the stationary blade *D* and with which it is discharged in the reverse direction upon the blade *E*, but the angle of this line is no longer 20 deg., and that of the relative entry R_e (see line *DE*, Fig. 12) approaches the bucket at much too broad an angle. The buckets are, therefore, so fashioned as to send the steam off at a sharper angle than that at which they receive it. Notice that a line drawn across the tips of a blade, as *mn*, Fig. 11, is not square with the line of the blade's movement. This results in a diagram more like that shown by the lighter lines in Fig. 12.

The final residual velocity V_3 is reduced to that indicated by the line *FG*. The initial energy is proportional to the area of the square on V_1, the energy of the steam as it enters the second rotating blade by the middle square, the difference being proportional to the energy taken out in the first blade. The residual energy is proportional to the smallest square and the energy taken out by the second blade to the difference between the areas of this and the middle square. The side of the second square is the V_2 and of the smallest square the V_3 of the lighter diagram. Unsymmetrical buckets on the moving wheel result in end thrust and must be used with discretion.

The reaction of a jet, or the force with which it pushes the nozzle backward, is *mass* × *velocity*. The absolute ve-

*$V_1^2 = (x + u)^2 + y_2 = x^2 + 2xu + u^2 + y^2$
$V_2^2 = (x - u)^2 + y^2 = x^2 - 2xu + u^2 + y^2$
Subtracting the two equations
$V_1^2 - V_2^2 = 4xu$
Hence $\frac{V_1^2 - V_2^2}{2} = 2xu = 2 \times x \times u$, which is the area ebhge, for eb = dc = 2x.

locity with which the jet leaves the blade is R_d in Fig. 13; the component of this velocity in the direction of the blade movement is $x = u + m$. The energy absorbed per second is the product of this force by the space moved through in a second, that is, by the blade velocity. It is, therefore, proportional to *mass* $\times u \times (u + m) =$ *mass* $\times (u^2 + um)$. The energy due to the issuing velocity is *mass* $\times \frac{V_1^2}{2}$, that due to the residual velocity *mass* $\times \frac{V_2^2}{2}$, and the difference *mass* $\times \left(\frac{V_1^2 - V_2^2}{2}\right)$.

Leaving off the mass, which is common to both, it would seem that, as in the case of Fig. 10, the rectangle $ebhge = u^2 + um$, representing the energy absorbed, ought to be equal to one-half the difference of the two squares representing to the same scale the initial and final energies. But the difference in the two squares is easily proven to be $u^2 + 2\,um$.* Twice the rectangle would be $2\,u^2 + 2\,um$. The difference in the squares is less than twice the rectangle by just u^2; that is, by just the square of the blade velocity. This is because there was in the steam not only the energy due to the velocity with which it issued from the jet, but the energy which it took to get it into the blade and moving with the blade's velocity. If a diagram similar to Fig. 10 be drawn with the line $ac = R_e$ at right angles to the movement of the blade, it will represent the action of a pure reaction turbine, and the additional energy will be found in the motion represented in the line *ab* of such diagram.

Staying a Furnace Arch

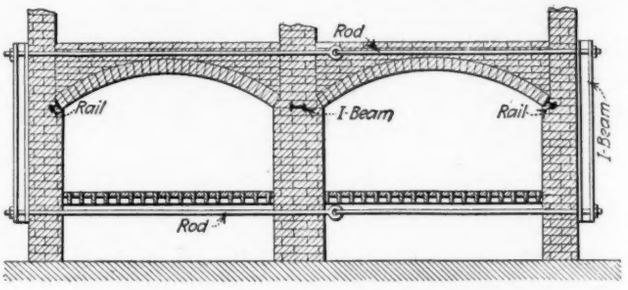
By J. C. Hawkins

Nearly every type of modern boiler furnace recommended to assist in giving complete combustion and to prevent smoke consists of firebrick piers, baffle walls, etc., placed directly in the path of the gases to cause a better mixture of the air and combustibles.

These gases are at a high temperature and impinge on the baffles and piers, often causing the firebrick to melt down quickly. The writer has seen bridge-walls in standard horizontal water-tube boiler settings, in which the top of the wall was melted and had run down on the back like huge icicles. The same trouble, only of a more serious nature, may occur in the arch of a dutch-oven furnace. If the arch is wide it exerts considerable pressure on the side walls and when the bricks become hot they tend to crush, the wall settles and exerts a greater pressure on the side walls. If the walls are not well stayed they will be pushed out and much air may leak in where it is not wanted.

The writer had considerable trouble of this kind in a battery of two vertical water-tube boilers with dutch-oven furnaces. The grate was 84 in. wide in each boiler and each furnace had a single-span arch. Each time the arch was renewed, which was about once a year, it pushed the walls farther out, and it was necessary to put in additional tie-rods. The accompanying sketch shows how this was accomplished. Four rods, 1¼ in. diameter each, were used; one was set at the end of the arch and one just inside the front wall. The lower rods were put in below the grate and close to it so as to be out of the way and not interfere with the pulling of the ashes. The top rods were covered with firebrick and ashes to protect them from the heat. The rods were made long enough to pass through the two settings, with a long thread and nut on each end. On account of close quarters the rods had to be linked in the center to get them in place. Four pieces of old railroad rail, each about 6 ft. long, were used as buckstays. In addition to these rods a 5-in. railroad rail about 8 ft. long was embedded in the center wall and also in the outside walls at the point where the arch rested, to strengthen the wall at this point.

This arch was constructed of a special grade of furnace brick and would last about a year, with hard firing. It is not usually possible to patch this arch when the center burns out, as it is generally sprung out of shape and sagged in the center. It is left as long as it will stand, then torn out and a new one built. In the parts of the lining where the brick is not subjected to the extreme heat common firebrick will be found satisfactory and cheaper than the high-grade brick used in the furnace. After the bricks become glazed over, which protects them, they should not be disturbed until it is necessary to repair them.



How the Furnace Arches Were Stayed

Fluid Cling-Surface

To overcome the necessity of heating Cling-Surface belt dressing without impairing its efficiency has long been the aim of the Cling-Surface Co., Buffalo, N. Y.

This has been accomplished, and the dressing has been converted to a semifluid ready for use at any temperature above 60 deg. F. In other words, no heating is required during eight to ten months of the year, nor in the winter if it is kept in a warm room. Where it is kept in a cold room or exposed to winter temperature, it need not be heated above 100 deg. F., and then only for a few minutes for softening.

U. S. Coal Production in 1914—According to Edward W. Parker, statistician of the United States Geological Survey, the total coal production of the United States in 1914 was about 510,000,000 short tons, a decrease of about 60,000,000 tons compared with the record output of 1913. Practically all of this decrease was in the output of the bituminous mines. The production of Pennsylvania anthracite in 1914 was not materially different from that of the preceding year, which was 81,718,680 long tons. In 1914, however, about 1,000,000 tons (principally nut and steam sizes) went into storage, so that the quantity sent to market was about 1,000,000 tons less than in 1913. The principal decreases in the production of bituminous coal were in the coking districts. It is estimated that in Pennsylvania alone the production of bituminous coal decreased between 20,000,000 and 25,000,000 tons, and that the larger part of this decrease was in Fayette and Westmoreland counties, which constitute the Connellsville and Lower Connellsville coking districts.

*$V_1^2 = y^2 + (u + m)^2 = y^2 + u^2 + 2\,um + m^2$
$V_2^2 = y^2 + m^2$
difference $= u^2 + 2\,um$

Interior Wiring for Lighting and Power Service--I

By A. L. Cook*

SYNOPSIS—The first of a series of articles covering the methods employed in making plans of lighting and power systems for industrial establishments and office buildings, and in calculating the sizes of wires required for such service. The treatment is such as to meet the requirements of superintendents or engineers in charge of such buildings, who may be called upon to make additions to or changes in the equipment; and no attempt has been made to cover problems which should be handled by the illuminating engineer or a power specialist. The first installment covers the voltages and systems employed, the National Electric Code Rules, the types, number and spacing of lamps and the determination of the lighting load.

The usual voltages employed for lighting are about 120 or 240 with a two-wire system and 120 for each side with a three-wire system. Either direct or alternating current may be used. Occasionally, three-phase or two-phase alternating current is employed for lighting, because of peculiarities in the conditions of supply. For alternating-current lighting 60 cycles is generally used, since 25 cycles is not as satisfactory owing to a flickering of the lights in some cases. It has been found, however, that tungsten lamps having a rating of 60 watts or more can be employed satisfactorily on 25 cycles. With ordinary inclosed arc lamps, 25 cycles is not satisfactory, although flame-carbon arc lamps can be used on this frequency. For direct-current motors, the standard voltages are 115, 230 or 550, and for alternating-current motors, 110, 220, 440 and 550 volts are commonly employed, although in some cases for very large motors, 2200 volts is used. The frequency may be either 60 or 25 cycles, and occasionally 40.

The voltages given for lighting and power service are the values at the lamps or motors. The standard generator voltages for direct current are 125, 250 and 600, and for alternating current, 120, 240, 480 and 600, which allows a reasonable drop between the generator and the load. In some cases a multivoltage system is used for motors, in order to give a ready means of varying the speed. This is not generally necessary, however, since modern direct-current motors permit wide speed variation by a change in the field strength.

The choice of a particular system for lighting or power service is affected by a number of factors, such as the character of the existing system or the central-station source of supply, and the relative sizes of the power and lighting loads. When an extension is to be made to an existing installation, the same system must be used for the extension, unless the addition is so large or the requirements differ so widely that a change in the system or the addition of a different kind of supply can be seriously considered. For a new plant more freedom of choice exists, and the relative merits of the various systems will therefore be considered.

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Direct vs. Alternating Current

For lighting, either alternating or direct current would, in general, be satisfactory, and the advantage of easy change of voltage in the case of the former makes it preferable in supplying buildings covering large areas. However, the lighting load is usually small, compared with the power load; hence the choice is fixed by the power requirements. The important advantages and disadvantages of alternating and direct current for power supply may be summarized as follows:

DIRECT CURRENT	ALTERNATING CURRENT
It is not generally feasible to use more than 240 volts for lighting. Therefore this limits the voltage of the system if supplied from the same generator as the motors.	The voltage can be easily transformed, using voltages suitable for lights and motors.
2. Maintenance is higher, owing to commutators.	2. There is no commutator; hence the motor is more rugged. It will stand larger momentary overloads, there is no danger of fire from sparks at the commutator and it is more reliable.
3. Wide speed variation of motor by simple means, with high efficiency.	3. Speed variation is difficult and the motor is less efficient at reduced speeds.
4. Motors have better starting characteristics for cranes and elevators.	4. Operation is not satisfactory on high-speed elevators and large cranes. Starting current is greater.
5. Starting current is lower for usual types of constant-speed motors.	5. Starting current for ordinary type is large. Special arrangements are necessary to reduce it.
	6. A somewhat larger generator is required for a given motor load.

The relative sizes of the power and lighting loads will have an important bearing upon the selection of the system. In some cases of light manufacturing, particularly if all the work is in one building, where the feeders would be short, direct current might well be used, employing 120 volts two-wire for small systems, and 240 volts three-wire, or possibly two-wire, for larger systems. If a two-wire system be used, the feeders would be about one-fourth as large for the 240 volts as for 120 volts; but, on the other hand, the lighting would have to be supplied at 240, which would entail somewhat greater cost for lamps and maintenance. It is better to operate the motors at 240 volts and supply the lights on a 120-240-volt three-wire system. By this means, the saving in size of feeders is nearly as great as if the entire load were supplied at 240 volts and the advantage of the lower-voltage lamps is secured. The additional power-house equipment is of small cost.

For most industrial uses, the alternating-current motor is satisfactory, and in some cases almost necessary, either because of the great distances from the power house or the

severe operating conditions due to dust, moisture, etc. Its principal disadvantage is the difficulty in adjusting the speed. With a direct-current system it is possible to obtain motors which will allow a speed change of three to one. When the speed is adjusted to a given value between these limits, it will remain practically constant regardless of the load. Such motors are extensively used for driving lathes and similar machine tools. It is possible to provide means by which the speed of an alternating-current motor can be adjusted to as wide a range as the direct-current motor, but usually at a sacrifice in efficiency; whereas, the direct-current motor has nearly the same efficiency at all speeds. Moreover, the variable-speed alternating-current motor, having been adjusted to a particular speed, will not maintain this as the load changes; instead, the speed will increase as the load decreases. This wide speed variation is objectionable where constant speed with varying load is necessary, as in machine-tool driving; but for some purposes, such as ventilating fans, centrifugal pumps, paper machines, and the like, where the load does not vary suddenly, the use of an alternating-current adjustable-speed motor is satisfactory. Alternating-current motors are not as satisfactory for cranes and elevators, owing principally to the difficulty of control, particularly when making stops. For this reason direct-current motors are to be preferred for high-speed elevators and large cranes. Therefore, in an office building where the elevator load is usually greater than the other motor load and the length of the feeders is not great, the direct-current system is preferable. For large buildings the three-wire, 240-volt system should be used, the motors operating at 240 volts and the lights at 120. Only in small buildings should the 120-volt two-wire system be used.

If the building is not supplied from a power plant on the premises, but obtains its supply from a central station, the type of service will depend upon the system of the supply company. If only alternating current is available it will be best to use alternating-current elevators unless the speed is high (above 300 ft. per min.) rather than provide the necessary transforming apparatus. For industrial establishments in general, the alternating current is to be preferred unless the cranes and variable-speed tools form a large proportion of the total load. If it is absolutely necessary to use direct current for some of the motors, it is better to provide alternating-current service for general uses, with a direct-current supply for cranes and special work.

When installing any wiring it is desirable to conform in all respects to the local rules governing such installations. The rules of the National Board of Fire Underwriters, called the "National Electric Code," form the basis of most of the regulations which have been issued by various cities and other parties interested, and must be followed in order to obtain fire insurance on property. These rules may be obtained gratis from the National Board of Fire Underwriters by applying to its New York, Boston or Chicago offices. The Inspection Department of the Associated Factory Mutual Fire Insurance Companies, with an office in Boston, has issued the "National Electric Code" with explanatory notes, thus giving in many cases more specific directions for the proper installation of electrical apparatus than is contained in the "Code." In many cases there are rules issued by the city inspection departments, which are substantially the same as the "National Electric Code," but care should be taken to see that the work not only meets the code requirements but also conforms to the local rules. In the following discussion the rules of the "National Electric Code" are followed.

Choice and Distribution of Lamps

The subject of the proper illumination of industrial establishments has in the past few years been given considerable attention on the part of factory superintendents and managers, who have begun to realize that it pays to provide sufficient illumination. Investigations have shown that an efficient lighting system increases the output from 2 to 10 per cent., and it has also been found that the number of accidents is materially reduced when adequate lighting is provided.

For interior illumination of buildings, there are available the following types of lamps:

	Lamp	Service
1.	Carbon-filament	A.C. or D.C.
2.	Gem- or metallized-filament	A.C. or D.C.
3.	Tantalum	A.C. or D.C.
4.	Tungsten, including "nitrogen" filled lamps	A.C. or D.C.
5.	Inclosed-carbon arc	A.C. or D.C.
6.	Metallic-flame or magnetite arc	D.C.
7.	Flame-carbon arc	A.C. or D.C.
8.	Nernst	A.C. or D.C.
9.	Cooper-Hewitt mercury arc	A.C. or D.C.

While all of the foregoing types have been used for interior illumination, the practice has now become so standardized as to make the tungsten lamp by far the most common for ordinary heights of ceilings. The metallic-flame arc and flame-carbon arc are used for lighting large floor areas with high ceilings, particularly where there is more or less smoke and gas. The so-called nitrogen-filled lamp, which is a special form of tungsten lamp with the bulb filled with nitrogen or a similar gas, is very useful where large lighting units can be employed, and the tendency is to use this in place of the metallic-flame or flame-carbon arc, owing to the reduced cost of maintenance. The mercury arc has also been used extensively, principally because of its small power consumption, but it produces such an objectionable color that it is unsuitable for many uses and can better be replaced by the nitrogen-filled lamp. This gives a light even whiter than the ordinary tungsten lamp with a power consumption not much greater than that of the mercury arc. Present practice, therefore, for rooms of ordinary height, has narrowed down to the use of tungsten lamps with glass or steel reflectors, mounted near the ceiling and arranged to give sufficient illumination to the entire room. In general, drop cords with individual lights have been eliminated as far as possible and are used only for special work which cannot be lighted from the overhead lamps. Where it is necessary to use individual lights, a 16-cp. carbon-filament or a 40-watt gem lamp is used. The latter is preferable as it gives the same candlepower as the carbon and requires about 20 per cent. less power. Table 1 gives data on the various sizes of tungsten lamps.

TABLE 1—DATA ON TUNGSTEN LAMPS*

Size, Rated Watts	Candle-power	Watts per Candle-power	Life, Hours	Approximate Current, Amperes 120 Volts	240 Volts
25	24	1.05	1000	0.21	0.11
40	39	1.03	1000	0.33	0.17
60	60	1.00	1000	0.50	0.25
100	105	0.95	1000	0.83	0.42
150	167	0.90	1000	1.25	0.62
250	278	0.90	1000	2.08	1.04
400	445	0.90	1000	3.33	1.67
500	555	0.90	1000	4.16	2.08
†200	222	0.90	1000	1.67	...
†300	353	0.85	1000	2.50	...
†400	534	0.75	1000	3.33	...
†500	714	0.70	1000	4.16	...
†750	1150	0.65	1000	6.25	...
†1000	1665	0.60	1000	8.33	...

*From figures supplied by the National Lamp Works of the General Electric Co. The above applies to 120-volt lamps; for 240-volt lamps the watts per candlepower are about 10 per cent. higher.

†Nitrogen-filled lamps of 120 volts only.

Sizes smaller than 25 watts are manufactured, but are not suitable for industrial lighting. For multiple inclosed-flame arcs the following values are typical:

	Direct Current	Alternating Current
Voltage	110	110
Watts	715	510
Amperes	6.5	7.5
Power factor	...	0.62
Candlepower	1740	1600
Watts per candlepower	0.41	0.32

In the case of the arc lamp, the candlepower refers to the average for the lower hemisphere of the lamp. For the tungsten lamps the candlepower and efficiency values are based on average candlepower in a horizontal direction when the lamp is vertical, no reflectors or shades being used.

It is not the intention to go into the details involved in the determination of the proper number and spacing of lamps for all classes of service, as this is a task for the illuminating engineer. Careful calculations of such a problem require considerable experience and a knowledge of the effect of reflection from walls and ceilings. It is frequently necessary, however, to make a rough estimate of the amount of power required for lighting, in order to provide the necessary feeder and generator capacity. There are a few simple rules that can be applied in such cases, which will give satisfactory results under usual conditions.

It is first necessary to determine the amount of power required for a given floor area. This will depend, of course, upon the amount of light necessary, which will vary with the character of the work carried on. Table 2 gives the number of watts required per square foot of floor area for different classes of work, with various arrangements of tungsten lamps. These values are based on good practice and will give first-class illumination under average conditions. The principal item which would affect these values is the color of the ceilings and walls. For offices, stores, corridors and drafting rooms it is assumed that both the ceilings and the walls are fairly light in color, while for factories, warehouses and power houses they would be darker and less light would be reflected. The figures given for general office illumination are sufficient for usual office work, while those for special illumination should be used where bookkeeping or work of a similar nature is carried on. The amount of power allowed for a drafting room is sufficient to provide suitable illumination without the use of individual lamps. For rooms where rough manufacturing is carried on and where close application to the work is not required, the figures for general factory illumination should be sufficient; for fine machine work, toolmaking and bench work, those for special factory illumination should be used. The lamps should be provided with suitable reflectors, in order to direct as much of the light as possible on the work. There is a great variety of these reflectors, but they can all be grouped in a few general classes, each of which is best adapted for particular conditions. There are on the market several types of glass reflectors which direct most of the light in a downward direction, but allow a certain amount to pass through to the ceiling. The best example of this type is the prismatic "Holophane." In order to have a good distribution of light, it is necessary to employ the proper style of reflector; hence a different size is manufactured for each size of tungsten lamp. It is necessary also to use the right type of shade holder in order that the lamp may be correctly located in the reflector.

Since modern systems of illumination are usually laid out to give practically uniform lighting over the entire floor area, it is necessary to use different types of reflectors for different heights of ceilings and spacings between lamps. The Holophane prismatic glass reflectors are made in three styles: "Extensive," for low ceilings; "intensive," for medium ceilings; and "focusing," for high ceilings. Glass reflectors are best adapted for offices, stores, drafting rooms and similar places, where it is desirable to light the walls and ceilings, as well as the work. They have also been used quite extensively for factory lighting, but are not suitable for use where there is danger of breakage.

Steel reflectors are made in a number of styles, with white porcelain-enamel surfaces, white painted surfaces, or aluminum painted surfaces. In general, the porcelain-enameled reflector is better than the others, owing to a great reflecting power, and the ease with which it can

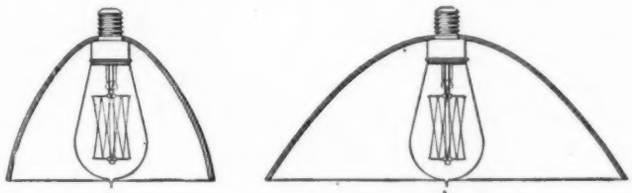


FIG. 1. BOWL TYPE DOME TYPE

be kept clean. There are two general types of steel reflectors—the bowl, shown in Fig. 1-*a*, and the dome, in Fig. 1-*b*. These reflectors are made in various sizes to suit particular tungsten lamps, and in various shapes for different heights of ceiling. The dome type (*b*) should be used generally; the bowl type (*a*), which incloses the lamp more than the dome, being used only when the lamps are mounted so low that they would be in the line of sight of the workmen. When steel reflectors are used, the ceilings are not illuminated, except by a small amount due to reflection from the benches or tables; but for many industrial applications this is not objectionable. In offices the steel reflectors do not give a pleasing effect. Values for either glass or steel reflectors are given in column A of Table 2, since they are both classed as direct illuminants. For the same character of walls and ceilings there would be only a slight difference in the amount of illumination produced by the two types.

TABLE 2—POWER REQUIRED FOR ILLUMINATION. TUNGSTEN LAMPS*

Class of Work	Watts per Square Foot Direct A	Indirect B
Office—general	1.00	1.60
Office—special	1.25	2.00
Drafting room	2.00	3.20
Corridors and halls	0.50	0.80
Factories—general	0.80	...
Factories—special	1.50	...
Warehouses	0.50	...
Stores	1.25	2.00
Power house	0.80	...
Storage	0.30	...

*If nitrogen-filled lamps are used, multiply the watts per square foot as given above by 0.75.

In some cases, particularly in drafting rooms, the indirect system of lighting is preferable. With this the light is directed upon the ceiling and is then reflected onto the work. It results in lower efficiency, but in many cases is justified, in order to eliminate troublesome shadows. A modification of this system involves the use of reflectors,

which allow a small portion of the light to be directed downward, giving what is called a semi-indirect system. A satisfactory arrangement with this system is to employ glass reflectors mounted on suitable fixtures and pointed toward the ceiling, instead of downward, as is usual. The indirect system depends for its efficiency upon light-colored ceilings and walls, and therefore is better adapted for use in offices, stores and drafting rooms than in factories.

The allowable watts per square foot for a given class of work can be found from Table 2, and when multiplied by the floor area, will give the total power required. It may seem to some that the height of the lamp above the work would have a decided effect upon the amount of power required, but this is not the case provided a suitable reflector and proper spacing of the lamps are employed. There is, however, a considerable difference in lighting depending upon the number of units employed and the color of the walls and ceiling.

As an example, the figures of Table 2 will be applied to the lighting of four floors of a factory building having a width of 46 ft. and a length of 135 ft., divided into nine bays 15 ft. wide, with a line of columns down the center of the building. Table 3 gives a tabulation showing the lighting to be provided for each floor.

TABLE 3—EXAMPLE OF LIGHTING CALCULATION

Floor	Character of Work	Ceiling Height, Feet	Area, Sq.Ft.	Assumed Watts per Sq.Ft.	Size of Unit	Actual Watts per Sq.Ft.
Basement	Storage	8	6210	0.30	60	0.35
First floor	Machine shop	14	6210	1.50	100	1.74
Second floor	Assembly	12	6210	1.50	100	1.74
Third floor	Stock room	12	6210	0.50	100	0.58

This building would employ direct lighting by means of tungsten lamps, and steel or glass reflectors. From the

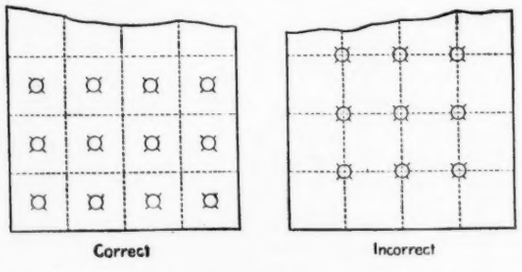


FIG. 2. SPACING OF CEILING OUTLETS

given floor areas and the allowable watts per square foot, the approximate amount of power can be estimated. This would be sufficient for an estimate of the total load required for the lighting, but in general it is best to choose the size of units and determine the number to be employed, since the spacing which must be used often modifies the total load.

The spacing and size of unit to be used are affected by the height of ceiling as well as by the arrangement of the beams or girders. There is a certain relation between the height of the lamps and their size, which must be adhered to as closely as possible, in order to get uniform illumination without objectionable shadows. For low ceilings the units should be small and closely spaced, while for high ceilings large units, more widely spaced, should be used. Table 4 will serve as a guide to the selection of the proper size of unit. This should be used in connection with Table 5, which gives the approximate spacing of lamps of different sizes.

The units should be mounted at least 8 ft. from the floor and more if possible; a height of 10 ft. being satisfactory for rooms with ceilings 11 to 16 ft. high. For higher ceilings, cranes and other obstructions usually fix the height of mounting. If deep girders divide the ceiling

TABLE 4—SIZES OF LIGHTING UNITS FOR VARIOUS MOUNTING HEIGHTS

Height of Unit above Floor	Size of Unit, Watts
Up to 9 ft.	40 or 60
9 to 11 ft.	60 or 100
11 to 16 ft.	100 or 150
16 to 20 ft.	150 or 250
20 ft. and above	250, 400, 500 and nitrogen-filled lamps or flame arcs

TABLE 5—APPROXIMATE SPACING DISTANCES FOR LIGHTING UNITS

Size of Units, Watts	Watts per Sq.Ft., Direct*	Spacing Distance	Size of Units, Watts	Watts per Sq.Ft., Direct*	Spacing Distance
40	0.3	11 ft. 6 in.	150	1.5	10 ft.
40	0.5	9 ft.	150	2.0	8 ft. 8 in.
40	0.8	7 ft.			
			250	0.3	29 ft.
60	0.3	14 ft. 2 in.	250	0.5	22 ft. 5 in.
60	0.5	11 ft.	250	0.8	17 ft. 8 in.
60	0.8	8 ft. 8 in.	250	1.0	15 ft. 10 in.
60	1.0	7 ft. 9 in.	250	1.25	14 ft. 1 in.
60	1.25	7 ft.	250	1.5	12 ft. 11 in.
60	1.5	6 ft. 4 in.	250	2.0	11 ft. 2 in.
100	0.5	14 ft.	400	0.8	22 ft. 5 in.
100	0.8	11 ft. 2 in.	400	1.0	20 ft.
100	1.0	10 ft.	400	1.25	17 ft. 11 in.
100	1.25	9 ft.	400	1.50	16 ft. 4 in.
100	1.5	8 ft. 2 in.	400	2.0	14 ft. 1 in.
100	2.0	7 ft.			
			500	0.8	25 ft.
150	0.5	17 ft. 4 in.	500	1.0	22 ft. 5 in.
150	0.8	13 ft. 8 in.	500	1.25	20 ft.
150	1.0	12 ft. 3 in.	500	1.50	18 ft. 3 in.
150	1.25	11 ft.	500	2.0	15 ft. 10 in.

*The figures given apply to ordinary tungst lamps. In general the spacing of lamps should be about 50 per cent. greater than their height above the work illuminated.

into bays, the lamps should be located slightly below the bottom edge of the girders if possible. Having fixed upon a suitable mounting height, a size of unit should be chosen by reference to Table 4. The rating of this unit divided into the total watts for the given floor area will give the required number of lights. This number should then be laid out upon a plan of the room and the spacing checked with the average values given in Table 5. The lamps should be located without reference to the individual machines, so that a change in the latter would not affect the system. Each light should, if possible, be located in the center of a square, the length of the side being the spacing distance assumed. The lamps should be arranged in parallel rows, the distance between rows each way being as nearly as possible equal to the given spacing distance. The distance from the wall to the first row should be about one-half the spacing distance, except where benches are located at the side walls, when the first row of lights should be located about 12 to 18 in. nearer the wall than the edge of the bench. If the room is divided into bays by deep girders or columns, each bay should be treated as far as possible as a unit, and the lights so spaced as to avoid shadows from the columns. If the size of lamp first selected does not give a suitable number for convenient location, a different size should be chosen and another arrangement tried. It is, of course, desirable to use as large a unit as possible, to reduce the cost of the wiring; on the other hand, a smaller unit gives more uniform distribution of the light, greater freedom from shadows, and less trouble due to one light being extinguished. With a smaller unit it is also possible to arrange a more flexible method of control, allowing some of the lamps to be extinguished during a part of the time, and resulting in a saving in power.

In the example selected, the basement requires about 0.3

watt per square foot. From Table 4 either 40- or 60-watt lamps could be used. From Table 5 it will be seen that 40-watt lamps, to give 0.3 watt per square foot, must be spaced on 11-ft. 6-in. centers. This does not work in well, since the bays are 15 ft. wide. If 60-watt lamps are selected the spacing could be 14 ft. 2 in., which would allow one lamp in each row per bay. Allowing four rows—two either side of the line of columns—gives a total of 36 lamps or $36 \times 60 = 2160$ watts, which gives $\frac{2160}{6210} = 0.35$ watt per square foot. The spacing of the rows would be $\frac{46}{4} = 11$ ft. 6 in., the two rows next the walls being 5 ft. 9 in. from the wall.

For the first floor about 1.5 watts per square foot will be required. From Table 5 it will be seen that a 100-watt unit would give a spacing of 8 ft. 2 in., and from Table 4 that this size is suitable for the height of ceiling. Therefore, two units per bay can be allowed, giving a spacing of 7 ft. 6 in. With six rows there would be a total of 108 units, requiring 10,800 watts. This is equivalent to 1.74 watts per square foot, which is somewhat more than was assumed. If the same number of 60-watt units were selected, a total of 6480 watts would be required, or 1.04 watts per square foot. Because of the columns through the center seven rows could not be used and with eight rows the spacing would be too small and the cost of installation too great. The distance between the wall and the first row would ordinarily be one-half the distance between the other rows; but in this case, as there would be benches along the walls, the rows next the walls could be located 2 ft. away and the other rows spaced evenly, giving about 8 ft. 5 in. for the distance between these rows. The other floors would be treated similarly. If there are no beams to divide the room into bays, the problem is simplified, but it must be remembered that the lamps should be located in the center of the square or rectangle and not at the corners; see Fig. 2.

Vacuum Heating Systems

By W. L. Durand

The use of vacuum heating systems has increased to such a large extent in the last few years that a general description of this method of heating may be of interest to engineers who are not familiar with it.

The advantages of a vacuum system over a gravity system may be summarized as better circulation, the use of smaller pipes and the absence of air valves. In buildings with steam power plants, when the exhaust steam is in excess of the requirements for heating, a vacuum heating system is a distinct factor for economy, since the back pressure on the engines can be reduced to atmospheric, thereby correspondingly decreasing the water rate of the engines, and for very high buildings, buildings with large floor areas, or a group of buildings, a vacuum system is practically the only type of steam heating system that will work satisfactorily.

One of the claims set forth by the manufacturers of vacuum valves is that wide variation in temperature is permissible. This is, however, more a talking point than anything else, as in the best-designed systems the pressure in the radiators is rarely lower than 2 in. of vacuum or more than two pounds above atmosphere, or a range of only about 10 deg.

There are three types of vacuum return valves on the market—the float, the thermostatic and the differential. The float type acts on the principle of a bucket trap, the condensed water raising a float which opens an outlet that allows the water to run out until just enough is left to maintain a seal, when the outlet is closed. A very small opening is left for the air to escape. This type of valve has the disadvantage that the opening for the escape of air allows steam to leak into the return lines, keeping the temperature of the returns so high that it is necessary to use an injection of cold water at the pump to maintain the desired vacuum.

In almost every case the manufacturers of the float type of valve have gone over to the thermostatic type. In this a hollow metal disk, usually made of copper, is filled with one or more liquids that vaporize at or around 200 deg. The action of this valve is extremely simple. Any air or condensed steam of a lower temperature than the valve is set for passes through the outlet, but as soon as steam reaches the disk the expansion of the liquid inside shuts the valve off. The advantage of this over the float type is that it is noiseless and does not pass steam, thus doing away with the use of jet water at the vacuum pump.

Thermostatic valves may be divided into two classes—one in which the expansion disk is on the pressure side of the valve and the other in which the disk is on the vacuum side. In most cases either kind of valve works satisfactorily, but experiments have shown that those with the disk on the pressure side can carry about 20 in. of vacuum, as against 10 in. for the other kind.

The third type of vacuum return valves is used on what is known as th differential system. In this a weighted check valve with restricted orifice is placed on the return side of each radiator. These valves are placed at different points, usually at the bottom of each return riser on the end of a horizontal run. The valve disk is weighted with a number of lead disks, the size of opening and weight of disks being so proportioned that for each lead disk a difference of pressure of 1 in. of mercury is required to lift the valve. By varying the number of disks any vacuum from 2 in. or 3 in. to 15 in. can be carried, as may be desired. This is the only type of valve by which the vacuum carried can be so varied, a feature which is advantageous for systems spread out over large areas or which have long horizontal runs.

In its installation a vacuum system is no different from a gravity system except that smaller pipes can be used, especially for the returns, and the vacuum valves are placed on the radiators and no air valves are used. The main return line is carried through a strainer to the vacuum pump, which can be either steam or electric driven, and from there it is pumped to a small tank with a vent open to the atmosphere. This relieves the tank of entrained air, and the return water is either fed direct to the boiler or through a feed-water heater.

It is not advisable to cover the returns in a vacuum system, as the exposed pipe surfaces allow the water to become sufficiently cooled for operation of the vacuum pump without the use of jet water. Where vacuum return valves are used dirt pockets should be provided, and in starting a new system the interiors of the valves should be removed and the system should be operated for four or five weeks as a gravity system.

The Peterson Power Plant Oil Filter

This filter, which is manufactured by the Richardson-Phenix Co., Milwaukee, Wis., embodies new principles of oil purification. Its operation is as follows:

The dirty oil enters through the strainer box at the top and passes down through the removable strainer, where large particles of foreign matter, such as waste and the like, are strained out; the oil then goes to the heating tray where its viscosity is reduced. It then flows to the compartment below the heating coils and down through the funnel. The further operation of the precipitation compartment is more clearly shown in Fig. 2.

Passing down through the tube conductor, the oil is spread out by a baffle under the lower tray. Under the action of the greater head which builds up in the tube conductor, the oil is forced to take a zigzag path upward, passing under and over several trays, as shown by the lines of flow. It then passes out through the opening below the heating tray to the filtering compartment. The separated water collects in the bottoms of the different trays and is bypassed to the bottom of the precipitation compartment by means of funnels that surround the tube conductor, and does not again come in contact with the traveling oil.

The water which is removed by the precipitation process is automatically ejected by an overflow tube at the right, which consists of two concentric pipes. The water flows upward through the outer tube and spills over the top of the funnel. The lower end of the tube can be connected to a sump or sewer. The funnel is threaded and can be raised or lowered, providing for proper adjustment for oils of different specific gravities. This water overflow simply operates on the U-tube principle; that is, the column of water in the outer pipe balances a column in the filter made up partly of oil and partly of water. As oil is lighter than water, the top of the overflow is a little lower than the level of oil in the precipitation compartment. As more water is precipitated out of the oil, the water level in the precipitation compartment tends to rise and the leg of the U-tube, which is inside the filter, becomes heavier because it is made up of a greater proportion of water and less of oil, thus water flows over the top of the funnel until the two legs of the U-tube again balance. In this way a low-water level is automatically maintained in the precipitation compartment.

Referring to Fig. 1, the level of the oil in the top tray is maintained constant by the skimmer, and the oil then flows through a pipe into the filtering compartment, which contains nine noncollapsible filtering units, such as is shown in Fig. 3. The oil passes from the outside to the inside of the filtering units, then out through the nozzles which project through the wall of the filtering compartment, to the clean-oil compartment. The nozzles on each unit fit into a spring-actuated valve so that any individual unit can be withdrawn and cleaned without interfering with the continuous operation of the filter. When the unit is withdrawn, this valve closes and prevents unfiltered oil from flowing into the clean-oil compartment. The filtering cloth is so arranged that it is free from folds or plaits, thus rendering every square inch active in filtering.

No oil can pass to the clean-oil compartment until the level in the filtering compartment reaches the outlets. Thus no filtering takes place until every square inch of cloth is submerged in oil; then as soon as a slight head builds up over the outlet, the process of filtration begins and is distributed over all of the surface, which is subjected to equal pressure.

The head of oil over the filtering disks is shown by an indicator at the top of the gage. When the filter is being operated at normal rating this gage should show a level of about three inches. If a greater height is in-

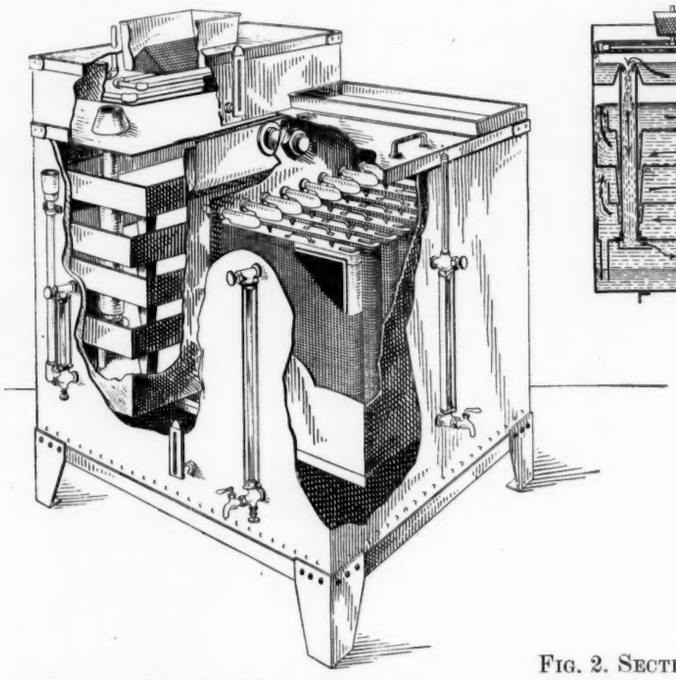


FIG. 1. SHOWING INTERIOR CONSTRUCTION OF THE FILTER

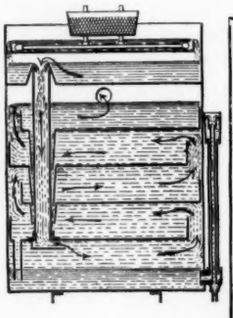


FIG. 2. SECTION THROUGH WATER-SEPARATING CHAMBER

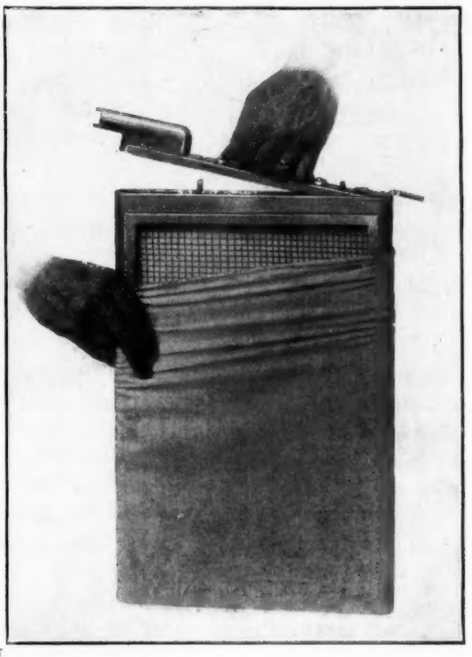


FIG. 3. NONCOLLAPSIBLE FILTERING UNIT

dicated it shows that the oil is not passing through the cloth as fast as it should and that the cloths need cleaning. The filters are rated at 3-in. head over the filtering disks, but space is provided for carrying a 6-in. head. Thus the apparatus is capable of handling short overloads of 100 per cent., so that in case a large batch of oil should be run in, the filter will be able to take care of it.

The advantage in arranging the filtering cloth in a vertical position and having the oil pass from the outside to the inside of the units is that the slime and sediment which collect on the cloth continually work toward the bottom and drop off, thus automatically tending to keep the surface clean. The filtering medium is a special grade of cloth that does not act as a screen, but actually filters the oil largely by capillary action.

The water level in the precipitation compartment should be carried as low as practical. The gage shows the clean-oil level. The cock on the fitting at the bottom of this gage provides for withdrawal of clean oil to use in cans for hand oiling. The right-hand gage shows the level of the oil in the filtering compartment. This should at all times be full of oil.

All of the level gages have sheet-metal guards in back of them which are white enamel on the inside. This makes it easy to see at a distance the oil level and also protects the glass from breakage.

A thermometer shows the temperature of the oil before it enters the precipitation compartment, thus enabling the engineer to adjust the quantity of heat supplied so that the proper viscosity will be maintained. Another thermometer shows the temperature of the oil in the clean-oil storage compartment.

The filter body is constructed of galvanized sheet steel reinforced with channel and angle iron. All joints are lapped and closely riveted and soldered.

The only parts needing periodical cleaning are the filter cloths. The filtering units can be easily removed without interfering with the continuous operation of the filter and should be lifted out and set in a pan of kerosene or gasoline and brushed down with a stiff brush; this is possible because all the sediment collects on the outside of the cloth. Occasionally, the cloths can be removed and washed in gasoline or kerosene to thoroughly clean them.

The filter is built on the unit principle, and has been constructed with a capacity of 7500 gal. per hour in a single unit.

Don'ts for Refrigerating Engineers

By Thomas G. Thurston

SYNOPSIS—Thirty-four sensible "don'ts" for the operating engineer handling refrigeration machinery.

Don't start an ammonia compressor without first noting that the discharge valve is open. The writer remembers when this was overlooked in starting a 250-ton vertical machine, and two men who were working up under the ceiling above the machine came near being overcome with the gas. If the machine is not provided with a relief valve, the head is liable to be blown off.

Don't forget to turn the water on the condenser just before or immediately after starting the machine. If this is not done until the head pressure becomes high, leaks are liable to start in the condenser, both from the excess pressure and the increased temperature. If allowed to go too long the results will be as bad as forgetting to open the discharge valve.

Don't neglect to watch the head pressure while starting. The header valve on a condenser had been shut, and the operator, in a hurry to start, forgot it and blew the relief valves. Watching the head pressure will warn you, if you have forgotten it, to turn the water on the condenser.

Don't be in too much of a hurry to open the suction valve when starting, especially if the machine has been shut down for a long time, without pumping down the expansion coils and suction line. Open the valve slowly and keep your hand on the discharge pipe if possible. If this suddenly gets cold and the discharge valves begin to work unusually quietly, there is liquid coming back with the gas and the suction valve must be choked off until the machine begins to warm up again, otherwise it may wreck the compressor and may cause loss of life. If there is a slamming, or pounding, in the cylinder similar to an engine getting a dose of water, shut the suction valve until it stops, as this is an indication of a dangerous condition.

Don't forget to close the suction valve when shutting down even for a little while, and don't forget to close the liquid valve in the liquid line from the receiver to the expansion coils, or to close the expansion valve when shutting down a machine, if it is working alone. If these are not done the liquid will accumulate in the expansion coils and the suction line and make trouble when starting up again, or until it is pumped out.

Don't forget to watch the stuffing-box when starting up. This usually has to be tightened when the machine has been shut down for a long time, and it must be let out again gradually as the rod and packing warm.

Don't try to run the packing after it gets burned or hard and loses its resiliency. It will wear the rod and waste ammonia. Packing is cheaper than new rods or ammonia.

Don't run the crossheads too loose on an ammonia compressor, and don't tighten the crosshead shoes any old way. Either will make trouble in the stuffing-box. Put the compressor crank on the crank-end center and adjust the shoes so that the piston rod is the same distance from the guides at both ends of the stroke.

On the compressor the thrust of the connecting-rod usually presses the crosshead against the top guide except when the crank passes the center, when the weight of the crosshead drops it down on the bottom guide. If the crosshead is loose this will cause the packing to leak and wear it out quickly. This applies only where the engine and compressor are placed parallel, or side by side; if they are placed tandem, or opposite each other, the thrust

on the compressor crosshead is the same as on the engine, and the guides should be adjusted accordingly.

Don't use more oil than necessary. If more is used it gets through the packing into the cylinder or goes out through the gas relief line into the suction pipe and the machine and out into the system, where it makes trouble.

Don't pump oil into the compressor cylinder. Usually enough oil leaks in through the stuffing-box and the gas relief line in addition to what circulates around the system to keep the compressor well lubricated. The only exception I have ever found to this was a 200-ton vertical double-acting machine. On this we had to pump about one-half pint every twenty-four hours into the top of the cylinders to keep the discharge valves working freely and to lubricate the cylinders.

Don't neglect to blow down the oil traps regularly. If you are not using more than a quart of oil every twenty-four hours, blow the traps at least twice a week, although every two or three days is better. If more than this is used, blow them at least every two days, and if the oil consumption is a gallon a day they should be blown every day.

Don't run the compressor excessively hot or ice cold; either one means loss of efficiency and capacity. The best results are had in the average plant by keeping the temperature of the discharge so hot that the hand can be held on it without burning.

Don't circulate water in the water jacket if the water is not warmer on leaving the jacket than on entering. To do so wastes water and refrigerating capacity.

Don't be satisfied with any suction pressure. Experiment with the expansion valves and see how high you can get the pressure without making the machine too cold; then try and keep it there unless the temperatures drop.

Don't run with a high head pressure unless it costs more for water to keep it down than for coal to pump against it.

Don't neglect to purge the air and foul gases out of the condenser regularly; this will help to keep the head pressure down. Any time the head pressure begins to climb without apparent reason or some of the coils get cold it is an indication of air or foul gas in the condenser, and the latter should be purged.

Don't neglect to pump out the air before starting in case any part of the system has been opened for alterations or repairs; it will save the trouble of purging it out of the condenser later on and save ammonia, as some ammonia always escapes with the air when purging.

Don't pump a vacuum on the system unnecessarily; it is likely to draw air into the system, and you will have to purge it out again.

Don't run with an insufficient charge of ammonia; keep enough in the system so that there is at least from four to six inches in the gage-glass on the ammonia receiver when running at maximum capacity. If the liquid level gets too low, some of the gas will pass over to the expansion coils with the liquid. Power has been used to compress this gas and water to cool it, and when it passes the expansion valve it will take some of the liquid that you have spent power and water to produce to cool it to the temperature of the suction gas. You are not only doing useless work in compressing and cooling the gas, but doing so absorbs some of the useful work already done, reducing the capacity and increasing the cost of operation.

Don't neglect the ammonia leaks. It does not take long for a dollar's worth of ammonia to leak out if there are a few small leaks in the system. Every week go over the points in the system with a lighted sulphur stick. The machine, and the high-pressure side of the system especially, need watching. Test the condensing water, jacket water and brine with litmus paper or Nessler's solution.

Don't neglect to shut the water off the condenser if it is shut down for any length of time, especially if it is old and subject to leaks; allowing the water to circulate over it cools it too quickly and may start a number of new leaks.

Don't neglect to clean the frost off direct expansion coils. The best way to do this is to run a hot gas connection from the discharge of the machine to the liquid line and pump hot gas into the coils; this will clean them quickly and thoroughly. Clean coils mean lower temperatures or the same temperatures at a reduced speed of the machine and therefore reduce the operating cost.

Don't neglect to inspect the valves and the false heads if there are any; see that they seat well, that the springs are not broken and are of the proper tension, and that there are no scored places on the seats and valves where the gas can blow through.

Don't neglect to pump out thoroughly before opening any part of the system for alterations or repairs; some men have a habit of letting whatever gas is in the machine blow to the four winds. It is just as easy to pump it out, much nicer to work around, and it saves the ammonia.

Don't break open an ammonia joint or any part of the system until you are certain the pressure is off, and don't be too quick in opening it even then. Loosen the bolts a little and open the joint easily. In case there should be pressure on it you can draw it up again before it gets too strong for one to stay around it. You will at least have a good chance to get away. Some men have lost their lives by neglecting to take this precaution.

Don't pull up too hard on a joint that is under pressure; you may break the flange, lose your life, and a part of the ammonia charge. If a joint does not stop leaking after it is drawn up reasonably tight, pump the line out and renew the gasket.

Don't get excited in case anyone gets a dose of liquid ammonia or ammonia-saturated oil; douse him with water. Turn the fire hose on him if there is one.

Don't connect a stop valve in an ammonia line so that the flow of the gas or liquid tends to close the valve. The writer remembers two cases where the valves came off the stems and caused much trouble. In one case the valve shut off the flow and blew the relief valve on the compressor. It took much time to locate the cause. In the other case the valve did not stop the flow, but hammered back and forth on the seat, wearing the disk and seat so much that a new valve was needed.

Don't neglect to keep the oil out of the expansion coils. If there are signs of oil in the system, open the coils as soon as they can be spared and blow them out, first with steam and then with air, and be sure they are thoroughly dry before closing them.

Don't open or shut a valve without first checking to make sure that it is the right valve.

Don't be in too much of a hurry when pumping an air-pressure test with an old machine on a system that has been in service for a long time. If there is oil in the machine or discharge line, it may cause an explosion due to the compressor getting hot enough to ignite the oil. Run the machine slowly and keep plenty of water on the water jacket if it has one. If the machine or the discharge line gets very hot, shut the machine down until it cools.

Don't forget to keep up a rapid circulation of water when pumping out a double pipe or a submerged condenser. Otherwise it will freeze and burst. The same also applies to a brine cooler.

Don't leave the suction lines uncovered outside of the coolers. It means extra work without anything to show for it.

Don't run an ammonia plant without an ammonia helmet. When something blows out some day it will save its cost in ammonia and may save life.

"Defender" Boiler-Room Appliances

The accompanying illustrations show the principal portable instruments recently developed by the Defender Automatic Regulator Co., Oriel Bldg., St. Louis, Mo. are made in two sizes, one for collecting over an 8-hr. period and the other over a 24-hr. interval. The collection chamber is tapered to compensate for the change in the rate of flow of the water from the chamber due to the decrease in static head as the water lowers in the vessel. Consequently, the gas is drawn in at a constant rate and an average sample for the entire period is obtained. A gage glass on the collector is provided to indicate the height of the water at any time.

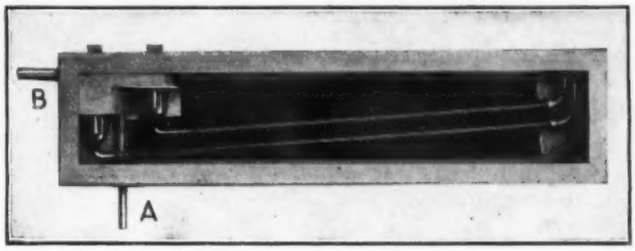


FIG. 4. DUPLEX DRAFT GAGE

Fig. 4 illustrates a "duplex" draft gage for indicating the draft in the furnace and at the damper or stack, and the drop in draft through the boiler. Outlet *A* connects with the lower or longer tube, which has a range up to 1 in. and is generally used for the uptake draft. Outlet *B* connects with the shorter tube, which has a range up to 0.9 in. to measure the furnace draft. The difference between the two readings gives the drop in draft through the boiler. These gages are furnished mounted in either an inclosed or open aluminum case.

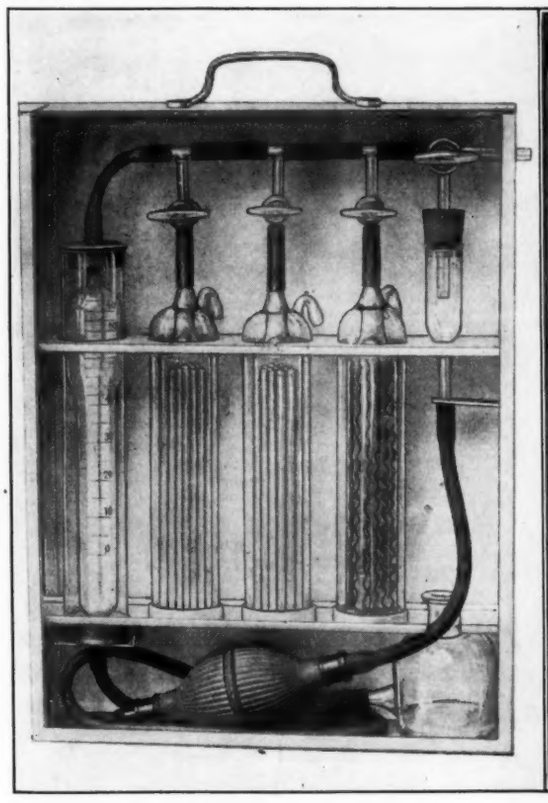


FIG. 1. THREE-PIPETTE MACHINE

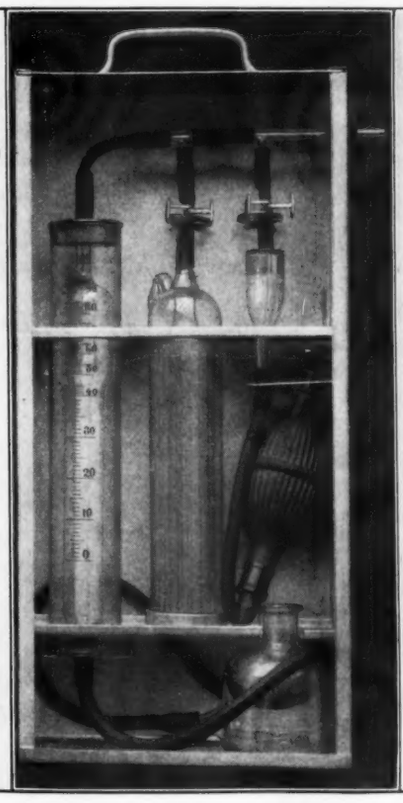


FIG. 2. SINGLE-PIPETTE MACHINE

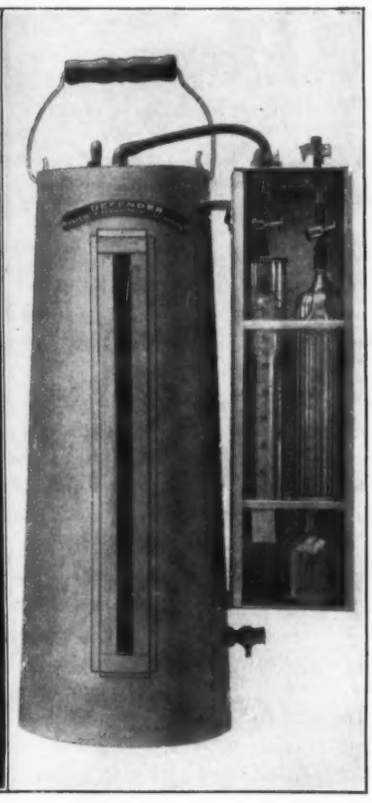


FIG. 3. GAS-SAMPLE COLLECTOR

Fig. 1 is a modified type of Orsat with three pipettes for determining the CO_2, O and CO contents of the flue gases. The case and covers are of metal, and the header, ordinarily of glass, is here made up of rubber connections and glass tees, not subject to breakage.

A single-pipette apparatus for use where the CO_2 content only of the gases is required is shown in Fig. 2. Fig. 3 is a combination gas-sample collector and CO_2 analyzer. Where desired the gas-sample collector is furnished without the analyzer attached. The collectors

In addition to the instruments illustrated the company also makes a high-range brass-inclosed thermometer for flue-gas temperature observations, single-tube draft gages and a "multiple" type draft gage mounted on a wooden panel for direct attachment to the boiler.

Saving in Federal Building Plant

Since its inception in 1905 the Federal Building in Chicago has been equipped with a boiler plant to supply steam for hydraulic elevator pumps, engine-driven air compressors serving a pneumatic-tube service, fan engines, boiler-feed and service pumps, and some live steam for the heating system. The exhaust from the units just mentioned only supplied about half the heat requirements of the building in the colder weather. Current for light and power had been purchased from the central station at a price of 1.9c. per kw.-hr. for the first 100,000 kw.-hr. per month, and 0.9c. for all current in excess of this amount. Eventually, it was decided to install a generating plant that would produce all the current needed and at the same time furnish more exhaust steam to the heating system. This equipment was described in the Apr. 28, 1914, issue of POWER, and from the saving made in the first month of operation it was estimated that the gross saving would exceed $14,000 a year. As shown in the accompanying table, this estimate has been exceeded by nearly $2000. In exact figures the gross saving effected by the plant during its first year of operation was $15,984.70. The total investment for the generating units and all additions to the plant was $43,000. Allowing 5 per cent. for depreciation and 3 per cent. for interest, which is more than the government usually receives, gives $3440 to be deducted from the total saving. This still leaves a balance of $12,544.70, which is over 29 per cent. on the investment and indicates that the plant will pay for itself in about 3½ years.

In the previous article the entire equipment was given in detail and in the following it is briefly summarized. The building is equipped with 45,000 sq.ft. of indirect and 65,000 sq.ft. of direct radiation on a two-pipe vacuum system. Seven passenger elevators of 3000 lb. capacity each, four 4000-lb. freight elevators and four 2000-lb. hydraulic lifts are served by one pumping engine, 16&20x 20x5¼x24 in., and two duplex tandem-compound pumps, 16&25x7⅜x18 in., working against a water pressure of 750 lb. For the pneumatic-tube mail-handling system there are four air compressors driven by cross-compound engines ranging in capacity from 75 to 125 hp. There is also a 20-ton absorption refrigerating system to cool the drinking water. To serve this equipment, five 350-hp. water-tube boilers had been installed. This gave plenty of reserve capacity for the generating plant. Incidentally, its installation did not increase the number of boilers in operation. There was plenty of unused space in the basement for the four generating units which were eventually installed. Two of these are 200-kw. machines and two 100-kw. generators directly driven by four-valve engines.

Steam flow meters are installed for the different services, and a water meter measures the total amount of boiler feed. The monthly and yearly totals, as read from these meters, are given in the table. With more exhaust steam to heat the feed water, it will be noticed that the average evaporation per pound of coal is 8.18 for 1914-15, as compared to 7.3 for the previous year. About the same amounts of steam went to the elevator pumps and to the air-compressor engines. The live steam to the boiler-feed pumps, fan engines, heating system, etc., was considerably less, as there was more exhaust available for the

OPERATING DATA OF FEDERAL BUILDING COVERING PERIOD OF TWO YEARS

Year	Month	Deg.-Hr. below 70 Deg.	Av. Temp. Feed Water from Heater, Deg. F.	Av. Temp. Feed Water from Econ., Deg. F.	Steam Supply: Av. Steam Pressure, Lb. Gage	Av. Evap. per Lb. Coal	To Elevator Pumps, Lb.	To Pneumatic Tube Compressor Engines, Lb.	To Pumps, Fans and Heating System, Lb.	Steam to Gen. Units, Lb.	Blow-down Water, Lb.	Total Water to Boiler, per Month, Lb.	Coal per Month, Long Tons	Power, Kw.-Hr. per Mo.	Light, Kw.-Hr. per Mo.	Total, Kw.-Hr. per Mo.	Cost of Current, per Mo.	Operating Expense	Saving Over Previous Year
1913	March	26,090	187	248	156.9	7.1	1,808,995	1,278,115	6,282,729	.....	129,115	9,498,954	587.75	26,130	84,210	110,340	$2,067.60	$6,019.07	
1913	April	15,336	179.9	244.9	156.2	7.1	1,778,109	1,171,217	3,851,117	.....	124,950	6,925,393	433.82	27,690	80,380	108,070	1,912.91	5,415.44	
1913	May	9,336	196.3	245.7	156.5	7.4	2,021,948	1,506,844	2,354,873	.....	129,115	6,012,780	352.8	35,340	79,220	114,560	2,007.26	5,033.12	
1913	June	3,932	197	246.6	157	7.8	1,679,797	1,779,345	1,374,225	.....	124,950	4,958,317	281.95	36,310	72,000	108,310	1,926.31	4,856.21	
1913	July	576	192.6	242.7	157.6	7.1	1,732,052	1,872,207	1,453,242	.....	103,292	5,160,793	315.95	39,090	70,790	109,880	2,036.87	5,117.06	
1913	Aug.	432	206.8	246.8	157.1	7.2	1,883,808	1,405,634	1,602,891	.....	77,469	4,969,802	302.35	41,560	70,010	111,570	1,936.32	5,025.96	
1913	Sept.	5,470	189.4	251	157	7.2	1,936,532	1,152,609	2,926,153	.....	126,616	6,141,910	336.8	61,440	72,310	133,750	2,054.89	5,173.53	
1913	Oct.	12,397	174.2	249.1	157.6	7.3	1,853,233	1,468,556	3,600,408	.....	103,292	7,025,519	419.95	74,280	84,940	159,220	2,190.06	5,795.16	
1913	Nov.	16,488	169.6	242.3	156.6	7.0	1,872,223	1,114,177	4,128,980	.....	99,360	7,215,340	456	65,560	82,380	147,940	2,221.30	5,708.30	
1913	Dec.	26,136	165.2	236.8	156.8	6.8	2,013,413	1,175,216	5,573,421	.....	103,292	8,865,342	576.04	54,000	93,990	147,990	2,270.78	6,353.60	
1914	Jan.	27,888	164.5	232.6	157.6	7.7	1,929,767	1,203,562	8,449,826	.....	103,292	11,686,447	673.85	46,080	93,180	139,260	2,206.90	6,575.83	
1914	Feb.	38,480	177.2	243.9	157.8	7.3	1,800,900	798,539	9,418,885	.....	93,296	12,111,630	733.72	36,970	81,810	118,780	2,026.55	6,685.70	
	Totals	183,061	183.3	244.2	157.1	7.3	22,310,777	5,926,061	51,016,750	.....	1,318,639	90,572,227	5470.98	544,450	965,220	1,509,670	$24,857.75	$67,760.98	
1914	March	25,384	182.2	248.8	154.5	6.6	2,045,846	970,535	7,625,645*	.....	103,292	10,745,318	713.0	45,460	89,880	135,340	.....	$4,519.96	$1,499.11
1914	April	15,624	193.2	256	148.9	7.3	1,743,781	924,626	7,264,331*	.....	99,360	10,032,698	611.6	53,360	82,730	136,090	.....	4,294.73	1,120.71
1914	May	6,936	195.7	259.4	149.7	7.5	1,665,266	946,364	1,659,421	5,069,864	103,292	9,444,207	556.1	37,850	86,440	124,290	.....	4,162.51	870.61
1914	June	2,688	202.4	261.4	150.6	7.0	1,906,813	871,935	1,573,619	3,969,756	99,960	8,422,083	527.2	36,580	73,470	110,050	.....	4,061.22	796.99
1914	July	202	202.2	261	151.9	7.2	1,849,427	1,010,533	1,307,654	4,184,630	103,292	8,455,536	524.3	34,050	71,890	105,940	.....	4,018.85	1,098.21
1914	Aug.	624	203.9	257.2	151.5	7.1	1,469,377	955,928	1,074,251	4,291,060	103,292	7,893,908	489.8	32,800	71,860	104,660	.....	3,985.92	1,040.04
1914	Sept.	3,192	201.6	254.6	152.4	8.5	1,763,252	1,585,416	1,074,251	4,333,440	99,960	9,356,319	492.5	31,480	75,190	106,670	.....	3,890.15	1,283.38
1914	Oct.	7,732	197.1	253.2	152.4	8.9	2,222,275	1,745,820	3,040,307	5,688,493	103,292	12,800,187	577.9	35,890	85,300	121,190	.....	4,341.50	1,453.66
1914	Nov.	18,816	165.5	234.6	151.7	8.3	2,036,190	1,522,902	2,365,018	5,188,156	99,960	11,232,226	607.8	33,100	79,680	112,780	.....	4,280.50	1,427.80
1914	Dec.	34,272	174.6	243	151.1	8.3	2,394,715	1,711,518	6,122,714	7,487,642	103,292	17,876,141	798.6	32,760	94,990	127,750	.....	4,899.01	1,454.59
1915	Jan.	32,256	181.2	244.8	150.7	9.3	2,509,650	1,576,644	7,459,192	6,995,082	103,292	18,673,860	825.6	31,760	92,970	124,730	.....	4,956.70	1,619.13
1915	Feb.	23,078	180.1	240.3	152	9.7	2,221,622	1,457,394	5,368,602	5,519,299	93,296	14,660,213	672.7	31,160	83,400	114,560	.....	4,365.23	2,320.47
	Totals	170,804	189.9	251.1	151.4	8.18	23,828,474	5,285,615	46,035,005	52,827,422	1,216,180	139,192,696	7397.1	436,250	987,800	1,424,050	.....	$51,776.28	$15,984.70

*Includes steam to generating units.

heating system. From the 1st of May to the end of February the engines used 52,827,422 lb. of steam. Dividing by the total kilowatt-hours generated in the same period, the consumption per kilowatt-hour is 45.8 lb. During the night hours the engines operated at light loads and the pressure was dropped to 140 lb., as compared to 160 lb. during the day. Besides, in the early part of the year all of the units were tuned up and more machines operated than were necessary. These factors account for a water rate which is higher than may be expected in succeeding years.

With the demands made by the other services about the same, the generating plant caused an additional operating expense of $8873.05, excluding the amount paid for central-station current. Dividing by the total output, the cost per kilowatt-hour amounted to 0.623c. The increase in coal consumption was 1926 tons, and to operate the generating units two extra men were required. The wages for these extra employees, coal at an average of $3.01 per ton and the additional oil and supplies required made up the excess in operating cost just referred to.

When the $24,857.75, the amount paid the central station for current, is added to the operating expenses of the plant, the total of $67,760.98 for the year ending February, 1914, exceeds the total expense for last year by $15,984.70. It is true that the current used during the year was less by 85,620 kw.-hr. More current was used for lighting, but the power demand was considerably reduced owing to changes in the service and to doubling up motors, making one do where formerly two had been employed. Had it been possible to reduce the electrical service in 1913 to the same figure, the amount paid for current would have been reduced $770.58, as the excess came under the 0.9c. rate. Putting both years on an equal basis, the gross saving would then be $15,214.12. Deducting the $3440 for interest and depreciation leaves a net balance of $11,774.12, which is 27.4 per cent. on the investment of $43,000.

Ratio of Circumferential to Longitudinal Stresses in Boiler Joints

By J. K. Linderhurst

SYNOPSIS—The article considers those conditions which sometimes make the ratio of the circumferential to the longitudinal stresses more than two to one.

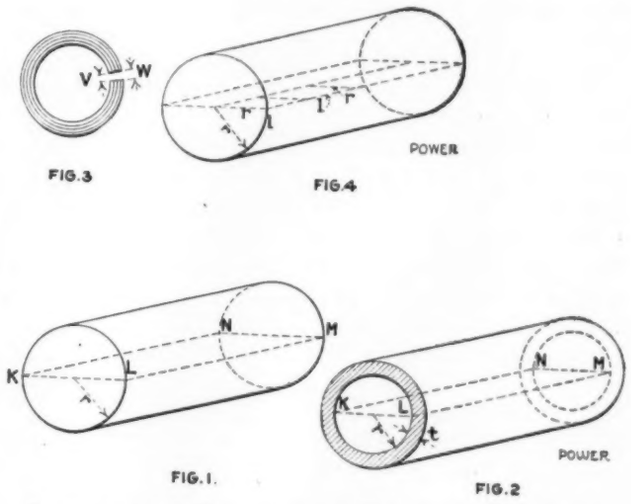
The average engineer who knows how to calculate the strength of boilers usually believes that estimating the strength of a cylinder to resist internal pressure is a simple problem. If all the factors are considered the problem is not simple, and the calculation of the correct efficiency of the longitudinal joint, usually considered the most difficult part of the problem, is one of the simplest.

Fig. 1, for example, is a part of a seamless cylinder, and if one were asked what is the relation between the circumferential and lengthwise stresses, he might state at once that the former is just twice the latter. The mathematical demonstration that this is so is apparently simple, for if the portion of this cylinder is taken of such length that the lines LM and KN equal the circumference of the cylinder, then the area on which pressure is exerted and tends to produce rupture is $2r \times LM$, when r is the radius of the cylinder.

The area exposed to pressure that will exert a lengthwise stress is the internal cross-sectional area of the cylinder, or $3.14r^2$. Now, since we have taken LM equal to $3.14r$, or one-half of the circumference of the cylinder, the first expression for area becomes $2r \times 3.14r$, or $6.28r^2$, which is twice the amount of the figure expressing the area of the internal cross-section, and therefore the lengthwise stress should be one-half of the circumferential.

No attention was given to thickness in making these calculations, for the shell was considered as a line without thickness. In Fig. 2, the thickness of the cylinder is indicated by the cross-section lines, and it is seen that if the circumference on the diameter KL is equal to LM and KN, the area of metal along LM and KN will not quite equal the cross-sectional area of the shell as indicated by the section lines, because this area equals the thickness times the mean circumference, which is halfway between the outer and inner surfaces, and not the thickness times the inner circumference.

Another factor not usually considered, and one which makes the difference between the longitudinal and maximum girthwise stresses greater than the one to two ratio, is as follows:



Illustrating Relation of Circumferential to Longitudinal Stresses in Boiler Joints

In considering the strength of cylinders, where the thickness is small as compared to the diameter, it is not customary to assume that the circumferential stress is unequally distributed over the thickness of the plate; but such is the case and, instead of the stress being equal throughout the plate thickness, it is greatest along the inner surface and least along the outer. No stress

within the elastic limit of a material can be applied without producing corresponding stretch. Since a cylinder cannot increase in diameter without stretching the inner surface a proportionately greater amount than the outer, the stresses in the plate will not be uniform. This is evident, if we consider a cylinder made up of concentric layers, as in Fig. 3. If such a cylinder were cut open to be free to expand, internal pressure applied to increase the diameter would cause the dimension W, which is the amount of separation of the outer layer, to be equal to V, if the layers of material were very thin. This illustrates how the material is supposed to behave in a solid plate cylinder under pressure. Since the actual stretch of all layers is the same, the stress produced in them is not uniform, on account of their varying length. The difference in the strength of a cylinder calculated in this way and one figured in the usual way is expressed by the ratio between the length of the outer and inner circumferences of the shell; or, since these circumferences are directly proportional to their radii, the ratio would be that between the inner and outer radii. For example, in Fig. 4, if t were of such dimensions that it would equal one-quarter of r (r being the inner radius of the cylinder), then the strength calculated in the usual way would be

$$\frac{thickness \times strength\ of\ shell\ per\ sq.in.}{inner\ radius} = bursting\ pres.$$

Assuming r at 4 in. and, therefore, t at 1 in., and 50,000 lb. per sq.in. as the strength of the shell material, we have, as bursting pressure,

$$\frac{1 \times 50{,}000}{4} = 12{,}500\ lb.$$

If the fact that the circumferential stress is not evenly distributed in the shell is taken into account, the bursting pressure is found to be

$$12{,}500 \times \frac{inner\ radius}{outer\ radius}, \text{ or } 12{,}500 \times \tfrac{4}{5} = 10{,}000\ lb.$$

bursting pressure, which is a difference of 20 per cent.

In boiler shells, where the diameter of the boiler is large, as compared to the thickness of plate, neglecting this feature in estimating the strength is not of importance, but to show the effect it may be well to consider an example. Take a boiler of 60-in. inside diameter and made of ½-in. plate subjected to a pressure of 125 lb. per sq.in., the boiler being assumed to be seamless. The load carried in a girthwise direction is

$$125 \times 30 \times 2 = 7500\ lb.,$$

but as this load is not equally distributed throughout the thickness of the plate, the maximum fiber stress at the inner surface is about 7620 lb. per sq.in. The endwise stress is due to the pressure of 125 lb. on an area of 2827.43 sq.in., or a load of 353,428.75 lb. The area of shell available to support this load is 47.4 sq.in., which is the cross-sectional area of the shell. Therefore, the stress per square inch longitudinally is

$$\frac{353{,}428.75}{47.4} = 7456\ lb.$$

so that the highest fiber stress girthwise is something over 2 per cent. greater than the lengthwise stress in the shell of this boiler when considering it as a cylinder without tubes. As has been stated, the slight difference in calculating boiler shells by either method is not enough to warrant consideration. It is well, however, to know just what we are talking about when we state that the girthwise stress in a cylinder subjected to internal pressure is twice the longitudinal stress.

Some like to express the relative values of these two stresses graphically and state that, as in Fig. 4, if we take any horizontal length of the shell, as l, and a corresponding length on the circumference l, the pressure on the triangle inclosed by the two radii produces the stress lengthwise of the shell on the section 1 in. long, and that the stress in the circumferential direction is due to the pressure on the rectangle lr. Since the area of this rectangle is just twice that of the triangle, the stress due to the pressure is in the same proportion.

The difficulty in this case is the same as in the case of Fig. 1 as regards accuracy, for only lines have been considered without thickness. If the metal to stand the load is to be equal in both cases, the length l in the case of the circumference of the shell would have to be taken on the mean circumference, in which case the triangular area would not be one-half of the rectangular, and therefore the resulting stresses would not be in the ratio of one to two.

This latter method of comparing the values of the stresses in a cylinder is called the graphic method.

Specifications for Steam Lines

By D. Craft

In specifications for steam lines for power using high pressures of steam, a mistake is commonly made in calling for lines as large as the connections of all the engines and turbine. It appears to the writer that many of the small engines and turbines used in power-plant work are provided with connections which are made large enough to supply the steam required at the minimum pressure likely to be used by any purchaser for the maximum rated capacity.

In our plant the pressure carried is 175 lb. All of the smaller machines here were originally connected to steam lines out of proportion to their capacity. For instance, two 5-hp. Terry turbines driving hotwell pumps had 1¼-in. steam lines; a 6x8x6-in. air compressor compressing to 50 lb. had a 2-in. line; two 6x9-in. engines driving circulating pumps and carrying a 6-hp. load had 2½-in. steam lines, and there were several similar instances of excessive sizes of connections.

We have reduced some of these lines to one-sixth of their original capacity. In addition to reducing losses from saving radiation we have saved considerable in expense and annoyance of keeping large throttle valves tight, and there is a marked reduction of governor trouble. The large governor valves throttled the steam so closely that from wiredrawing of the steam the valves required frequent grinding, reseating and renewing.

By reducing the size of these valves and lines we have overcome our trouble. Where we made these reductions in pipe sizes the lines were made only large enough to transmit 30 lb. of steam per hour for each horsepower required, with an allowance of about 10 per cent. for drop in pressure.

The Most Economical Pipe Size for the average demand should be used, with provision for increasing the boiler pressure or opening a supplementary "booster" line during maximum demand when the engine cannot otherwise carry the load. Recent tests here and abroad prove that substantial saving may follow a reduction in pipe size or shutting off booster lines, thereby saving in condensation and radiation losses.

Editorials

Factors in the Engineer's Salary

What determines the contents of the engineer's pay envelope? Surely, in too many cases his salary, or whatever his remuneration may be called, is the result of custom on the part of the establishment which he serves—a sort of haphazard adherence to the "going" rate of pay in the local community in this occupation. Surely, there are other standards by which employers may fairly gage the worth of their engineers—standards which take into account their responsibilities, their service performance and powers of constructive suggestion.

Because a ten-thousand-kilowatt turbine today requires less attendance than a two-thousand-five-hundred-kilowatt engine-driven unit did a dozen years ago, some people doubtless think that the engineer has an easier time in a typically modern plant, and therefore is not entitled to higher compensation. Again, some hold that because a plant is put on an eight-hour shift where a ten- or twelve-hour schedule was formerly maintained, the engineer's duties are correspondingly lightened and he deserves little consideration in the direction of "raises."

If any single factor is to be picked out as paramount in rate-making for operating engineers, that factor should be responsibility. The value of the service rendered by the plant is one great test, and the value of the equipment in the engineer's charge is another. It matters little whether the machinery is highly automatic in its operation so far as engineering responsibility is concerned, compared with the amount of money invested in it and the penalties of service interruptions. There may be less manual effort in handling a fifteen-thousand-kilowatt turbine and its auxiliaries than in operating an engine-driven outfit of a fifth that capacity, but the risk of damage, the question of daily cost when such a machine stands idle, the importance of its output, and the value of technical judgment in inspection and maintenance, all tend to place the man in charge of high-powered units in a special class as regards payment—a class which is recognized in many concerns, but which on principle ought to be appreciated all along the line more than it is.

In a word, the hours of daily service should cut little figure in salary or wage determination, and the same may be said of the output per unit. At first blush, the plant turning out a big total of daily horsepower-hours might be set up as a standard of payment, but while peak loads increase the engineer's anxieties, the daily test of his work comes down to his ability to turn out whatever output the load demands, at the lowest cost consistent with reliable service. The ability to save the plant owner money, realized in the daily work of the engineer, should not go unrewarded. Profit sharing is just as good a plan in the power house as in the factory, but a good many employers have yet to realize it. Certainly, the engineer who works diligently to improve the efficiency of his installation, who studies how to make it yield the best possible service, and who knows from accurate records just what the physical results of those efforts are, is demonstrating his fitness for responsibility and is keeping his "cutting edge" sharp to good purpose.

Analysis of local conditions in power plants should accompany consideration of the problems of compensation in the future, and the farther the plant owner gets from mere imitation of what others are doing, the better it will be for all concerned.

Art and the Steel Stack

A chimney has two principal functions—one to produce a draft, the other to take the gases high enough above our heads so that we do not notice them. Until someone invents an invisible chimney we shall have to put up with tall tubes sticking up into the sky line.

Various attempts have been made to disguise chimneys in the interest of the artistic. Greek columns, campaniles and other architectural units have been pressed into service, but the incongruity of such works of art belching clouds of smoke has made them less artistic than the chimney which pretends to be nothing but a chimney. Brick and concrete afford materials which can be made more or less ornamental—the former by shape and color, the latter by shape alone. The steel stack seems seldom to be anything but a parallel tube without ornamentation.

A steel stack has very decided advantages from a financial point of view. If it had to be replaced every year it would cost only about what the interest would be on a brick stack and its foundation. As it lasts from five to twenty-five years, according to the coal and weather, there is little to be said against it except as a matter of looks. There is a prejudice on the ground that it will not draw, but that is probably ill-founded. If there were any great difficulty from radiation it could be helped by means of a jacket or an inner tube, neither of which is often resorted to. On the score of looks there is nothing that can be said in favor of the ordinary factory stack. A degree of ornamentation is sometimes attempted by painting colored bands or by lettering the name of the owner on the side, but that is bad in that it attracts attention to an ugly thing.

It would seem as though a steel stack could be designed the lines of which would be at least pleasing when seen from sufficient distance so that the material was not in evidence. There is no insurmountable difficulty in making a tapering tube of steel. It is even possible to give it the slight swelling which appears to be necessary to make it look like a straight taper. The largest expense appears to be for an ornamental head, which may cost as much as the rest of the stack, but which, if made on simple lines relying on distance to obscure the lack of detail, should not be very expensive. Then if, instead of painting it a dead black, it could be made a gray or some neutral color, it would be still less noticeable, and that is the real object to be attained—practical invisibility, or making it attract as little notice as possible.

Looking Out for the Fine Points

The small details in the layout of a power plant, although often receiving scant attention, are important factors in its operation; and it is surprising how nearly every plant illustrates practice which is either worth following or which might well be avoided. Station lighting, for instance, too often receives insufficient consideration, particularly in the use of shades or reflectors capable of concentrating the light on machine parts needing frequent inspection or adjustment. Also, the use of unshaded lamps mounted low on columns in stations with high ceilings is extremely wasteful. Boiler rooms are notoriously ill-lighted in many stations, and here is a field for the practice of engineering skill along lines as yet relatively undeveloped. No lighting installation, however, will give adequate results if neglected as to periodical cleaning of fixtures.

The numbering of switches and motor starters to correspond with the apparatus controlled has important bearing upon convenience of operation. In emergencies requiring the sudden stopping of a motor-driven centrifugal pump and the starting of another, no time should be lost through the manipulation of the wrong switch. Similarly, the labeling of lighting switches with appropriately keyed circuit numbers, including specific areas covered, when feasible, is desirable. The mounting of generator field rheostats sometimes is a troublesome problem. In a recently completed station this apparatus was placed behind the switchboard, but so far above the floor as to obstruct the light from the windows, consequently it had to be blocked up with a wooden strut which obstructed the passage at the rear of the board. In a section of the plant containing refrigerating coils, lights were placed on the alternate stairway landings only, thus giving too dim an illumination for rapid and safe travel, and in the freezing room no effort was made to protect the local switches and fuses from vapors.

Although many of these loose ends are taken in hand after the plant is in regular operation, they are usually sources of inconvenience, extra cost, and danger, and should be guarded against wherever possible.

⁂

Making the Dollars Produce

Frequently, the operating engineer of a factory power plant is called upon to make changes or extensions to the existing plant, possibly occasioned by the addition of building or machinery made necessary by the desire of the owner for increased facility. At such times the importance of intimate knowledge of details of the existing power plant, together with the exact knowledge on the part of the operating engineer of the actual value utilized of the owner's dollar's worth of coal, cannot be too forcibly insisted upon.

The operating engineer who is content to just operate his plant without knowing exactly what part of the owner's dollar invested in the coal pile is being put to useful purpose, will fail when called upon for advice to meet the new conditions. He is the one who will sulk in the corner when the boss employs outside talent to solve his problems. In the corner with his face to the wall he will remain, for he has entirely neglected to grasp and use the opportunities given him.

How many operating engineers have traced the pound of coal fed to the boiler from the pile to the switchboard? How many ever stop to realize that only about fifteen cents out of a dollar's worth of coal actually reaches the switchboard? The missing eighty-five cents in too many instances is almost a total loss, whereas with proper knowledge and equipment as much as sixty cents of this can be saved and put to proper use.

The salesman who sold the engine and generator boasts about the combined efficiency of his apparatus being 95 per cent. This is not to say that the owner is getting ninety-five cents at the terminals of the generator on the dollar paid for coal. He is not. This combined efficiency means that for every hundred pounds of steam given the engine at the throttle, all the electrical energy that can be obtained from ninety-five pounds is realized at the terminals.

Possibly one will ask, why try to save the exhaust steam from the engine? The answer is that the steam exhausting from the engine contains more than twice as much heat value as is actually used in the engine and transformed into electrical energy.

The exhaust steam should be turned to good use, such as heating the buildings, or possibly supplying apparatus required in the process of manufacture, such as drying rooms and kettles, and also to heat the feed water supplied to the boiler, thereby reducing the work required from the coal.

Until recently, the owner invariably looked upon the coal bill as a necessary evil. Times are changing. The efficiency engineer is making rapid progress. Be in a position to tell the boss just where his dollar's worth of coal is going. Maybe, when he asks your advice on new equipment you can tell him that by making certain changes in the existing plant, there will be boiler capacity enough to carry the increased load, thus materially lightening the burdens on his pocketbook.

Don't wait for the efficiency engineer to tell your boss where his money goes; get there before him. You are the one that should tell him.

⁂

Is it any more onerous or less reasonable to require a user of a boiler to provide it with an adequate safety valve, than to require him to furnish an adequate fire pump; to insist upon an open exit from a fire-room which is liable to become a torture chamber filled with scalding steam from the bursting of a steam pipe or fitting or of a boiler tube, than from an ordinary workroom; to require the use of furnace doors which will not allow the fire to be blown out over the premises and the people, and to provide an outward escape for the steam and gas from a boiler setting in case of a bursting tube, than to insist upon guards around belts and flywheels? Has not "safety first" as much import in the power plant as elsewhere?

⁂

The head of an organization of steam-boiler firemen in New Jersey recently informed us that the educational work that has been vigorously conducted among engineers in that state during the past few years has had the effect of establishing unusually favorable relations between engineers and firemen. There is more thorough coöperation between the two today than ever before, he stated.

⁂

When the average man begins to educate himself he soon learns that he does not know as much about his calling as he thought he did. Once he is conscious of this, he is likely to be less arrogant and more helpful to his subordinates.

Correspondence

Maximum Lift of Steam Ejector

Referring to the request by Charles S. Palmer, as printed on page 481 of the Apr. 6 issue of POWER, and in particular to the question relative to the maximum lift that the steam ejector when applied to priming centrifugal pumps can be expected to develop, I offer the following:

The depth in feet from which the water can be raised to the pump by vacuum created by the ejector depends upon local conditions, such as tightness of piping and pump parts and the steam pressure used. Theoretically, it should be 33+ ft., which would indicate perfect vacuum, but in practice 25 ft. is near the maximum when the best range of steam pressures—40 to 80 lb. gage—is employed, and all valve stems, stuffing-boxes, pump and pipe joints are tight against air leakage. Usually considerable care must be exercised to keep the air leakage down to a minimum.

With Mr. Palmer, I should be much pleased to receive through your columns the common-sense explanation of the collapse of the discharge pipe on the pump mentioned in the article referred to.

PENBERTHY INJECTOR CO.,
L. A. PURCELL.

Detroit, Mich.

Priming a Centrifugal Pump

Some years ago I had the pleasure—and the work—of spending a season at a fairly large-sized irrigation plant in the cane and rice belt of southern Texas. I do not remember the capacity of the pumps, but it was considerable. There were two centrifugal pumps, each driven by a tandem-compound Corliss engine of the nonreleasing valve-gear type; the speed was 175 r.p.m. Each had a separate suction and discharge, the former being 72 in. and the latter 45 in. diameter.

These two pumps formed one of the two plants that were necessary to put water on the rice lands. The plant at which I was stationed was called the lower lift. It was on the Brazos River and the engine-room floor was about 50 ft. below the surrounding land level. This was necessary, as the water level in the river became low during the dry season, and then the pumps had about 12-ft. lift.

From the river the water was discharged into a canal about 35 ft. above the pumps. At low water this gave a total lift of something like 47 ft. The shaft stuffing-box was, of course, water-sealed. These pumps would easily lose their suction if they slowed down as little as five revolutions per minute below their regular speed. They always had to be primed when the water in the river was lower than the engine shaft. There was no valve in the suction line. The discharge line was provided with a gate valve operated by hydraulic pressure, which was usually assisted by a half-dozen men with chain blocks and bars.

There was a 4-in. steam ejector connected at the highest point in the pump case. Whenever a pump lost its suction it was stopped, the discharge valve closed and the ejector started. Sometimes the air would be dispelled and the case filled with water in from 15 to 20 minutes, but generally it took over half an hour and sometimes longer, owing to leaks in the pump and piping. The ejector took more steam than the engine and the boilers were crowded during priming. Often, while priming one pump the steam pressure would become so low that the other would slow down and also lose its suction. There were times when the ejector would just begin to discharge water when it would have to be shut off owing to the danger of the other pump slowing down.

As soon as the ejector began discharging a full stream the pump was started and brought up to speed as quickly as possible. The discharge valve was never opened till the pump was up to speed and then opened slowly. It was inconvenient to start either of these pumps with the discharge valve open. Although the ejector was a veritable steam eater it was the best means of priming we could get.

The other plant of this irrigation system was located about 19 miles from the river and took water out of the canal and discharged it into the irrigation ditches which supplied the rice fields. This plant was known as the upper lift and the pumps were built on the ground level, so there was scarcely any suction lift.

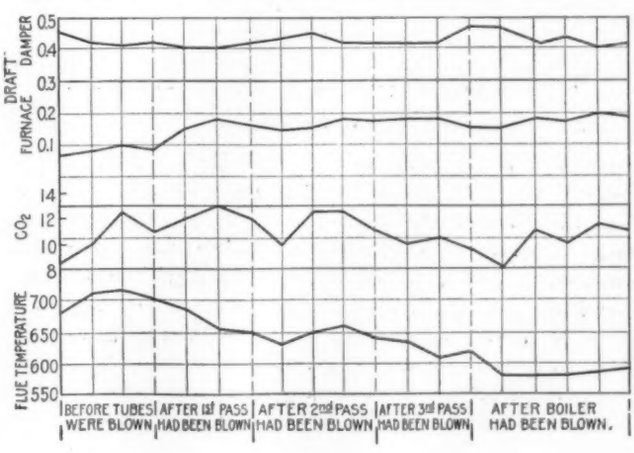
A. G. SOLOMON.

Chicago, Ill.

Soot Removal

In the Feb. 16 issue I read with interest the editorial "Soot," page 238.

Some time ago I made some observations on a horizontal, vertical-pass, water-tube boiler to determine the effect



EFFECT OF SOOT REMOVAL

of soot removal on the combustion and the temperature of the gases just after leaving the boiler.

It is our custom to blow the external surfaces of the tubes off with compressed air at about 100 lb. pressure. This is done daily except Sunday, which means that the

boiler is accumulating soot and dust for 48 hours. The accompanying curves were plotted from data taken after each pass had been blown.

The draft pressures and condition of the fire were kept as nearly uniform as possible. The boiler is served by a front-feed inclined grate stoker. The temperature was taken at a point just after the gases left the last pass. The flue-gas sample was taken at the same place, and analyzed with an Orsat.

It will be interesting to note in these curves the peculiar relation between the draft pressures and CO_2.

H. R. Blessing.

Philadelphia, Penn.

The editorial in a recent issue of Power relative to the soot problem furnishes food for thought. Why is it that many power plants equipped with flow meters, CO_2 recorders, pyrometers, draft gages and other appliances designed to promote boiler efficiency still stick to ancient hit-or-miss methods of removing soot deposits from their boilers? In Germany power-plant operators realize that soot has about five times the heat-resisting qualities of asbestos, and most boilers have stationary soot-blowing equipment.

In this vicinity such installations are, comparatively speaking, few. The writer has in mind a plant where boiler tubes are blown twice a week. After this blowing, from 1½ to 2 tons of soot is removed from each boiler. The work is done by an attendant with a hand hose, and as such men are but human, and dragging a hot steam hose about a boiler room is anything but a pleasant job, more than one dust slide is skipped in the operation. The time saved between the opening of a couple of valves on a stationary equipment and the shifting about of a hand hose is so apparent that no comment need be made on it.

There have been numerous interesting and instructive discussions in Power on flue-gas analysis, boiler settings and general power-plant practice, but this important subject of soot removal has not received much attention. Will not some who have equipment of this kind let us hear about it?

J. Priefer.

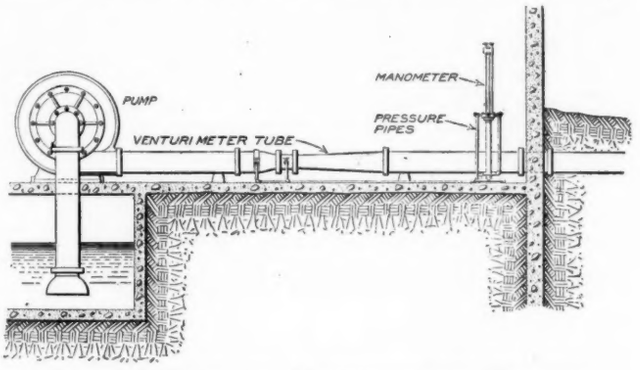
Brooklyn, N. Y.

Testing Small Centrifugal Pumps

I read with a great deal of interest the article by M. R. Blish in the Mar. 16 issue. Mr. Blish devotes considerable attention to the measurement of the discharge by means of weirs. I would like to suggest, however, that a much more convenient way of measuring the output from centrifugal pumps of either small or large size is by the use of a venturi meter tube with a mercury manometer. Such an arrangement is shown herewith. The tube is placed directly on the discharge from the pump and the pump performance may be tested under actual operating conditions if desired. This arrangement also has the advantage of being permanent if so desired, and a daily record of the pumpage may be kept. Any falling off in performance may be noted as soon as it occurs and the trouble corrected, if possible. If it is desired to test the pump under a greater range of heads than could be obtained at any particular time, the arrangement can be modified somewhat by placing a valve directly on the outlet cone and throttling it to obtain the head desired.

Mr. Blish in his article describes the use of a baffle plate to prevent serious velocity of approach of the water to the weir. With the Venturi tube this will not be required since the method of measurement by taking pressures at the upstream and throat or constricted portions of the tube is independent of the initial velocity of the flowing water.

The instrument is also considerably more convenient to read than the hook gage used with the weir. The manometer shown is a modification of the U-tube type and employs the same principle as the single-tube mercury barometer, the heavy pressure from the inlet of the meter tube being received upon the surface of the mercury in the



Venturi Tube and Manometer for Testing Centrifugal Pumps

large well at the bottom of the instrument. The throat pressure is conducted to the top of the single glass tube. The scale may be graduated directly in gallons per minute. This type of instrument makes unnecessary the adjusting of the scale or rod and is particularly convenient in cases where it is desired to take frequent readings of suction and discharge heads, speed, temperature, etc., at frequent intervals with a small number of observers.

Alan A. Wood.

Providence, R. I.

Feeding Graphite to Boilers

The writer has experimented for several months to discover a satisfactory method of feeding graphite into stationary boilers and has found the following to be best. The feeder shown in Fig. 1 is used in plants where compressed air is available, and that in Fig. 2 where compressed air cannot be had.

Fig. 1 is composed of a 6-gal. galvanized-iron tank *A* with a loose-fitting lid on top; *B* is a 1-in. pipe extending through the bottom and three-fourths of the way to the top of the tank, with a tee on top to prevent the attendant from filling the pipe with graphite when he is charging the tank. A 1-in. gate valve or straight-way air-cock *C* is used as shown. A globe valve should never be used here on account of its liability to clog. A 1-in. line *D* should be connected to the suction of the feed pump. A sight-feed *E*, through which the attendant can see the amount of graphite being fed, is provided.

A ¼-in. line *F* should be connected to the city water line or to the discharge of the feed pump. A nozzle *G* is used with a ⅛-in. opening pointing to the bottom of the tank on a 45-deg. angle, to prevent graphite from

settling to the bottom of the tank and keep it well mixed with water. A ½-in. valve *I* is used to drain the tank for recharging.

This feeder should be installed, so that the sight-feed is above the water line of the heater, and so that it will not fill with water.

To charge, close the valves *H* and *C*, open *I*, and drain water into the bucket. If this water contains graphite it

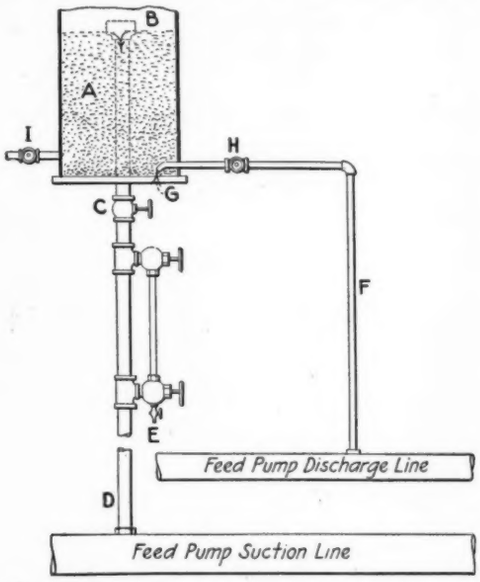


FIG. 1. GRAPHITE FEEDER USED WHERE COMPRESSED AIR IS AVAILABLE

can be put back into the tank after charging. After putting in the required amount of graphite, pour the water drawn from the tank back into it, stirring the contents well to mix thoroughly, open valve *C* wide and regulate the feed with *H*.

Fig. 2 consists chiefly of a 150-gal. steel tank *A*, four ⅜-in. pipes *B* with $\frac{1}{32}$-in. holes drilled 2 in. apart on one

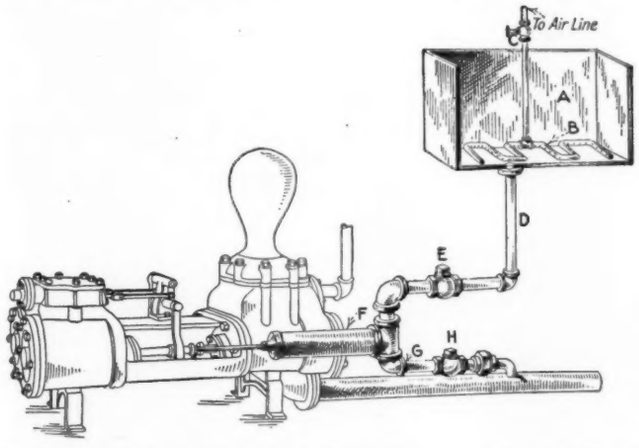


FIG. 2. GRAPHITE PUMP OPERATED BY BOILER-FEED PUMP

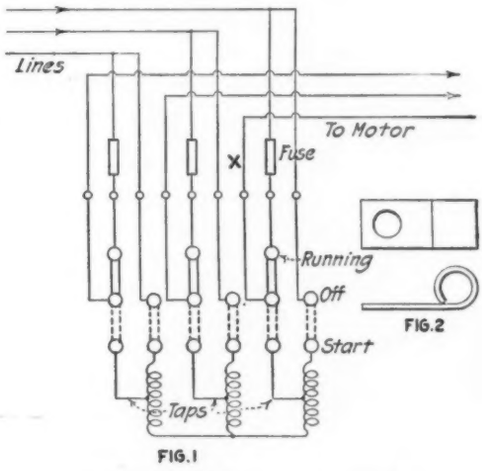
side and arranged in the tank so that the holes point downward, to keep the graphite from settling. A ½-in. air line *C* is connected to the compressed-air supply; *D* is a 1½-in. suction line; *E* 1½-in. swinging check valve; *F* a single-acting pump made of 2-in. brass pipe with a 2x1½x1-in. pipe tee on one end and the bonnet of a 2x2½-in. valve on the other. The piston head is made of brass, with two packing rings. The piston rod is made of ⅜-in. cold-rolled brass, with a permanent nut on the end of the rod and a sliding nut with a setscrew, so that the stroke of the pump can be adjusted to any desired length, regulating the amount of graphite fed with each stroke of the feed pump. The graphite pump is driven by the piston rod of the feed pump; *H* is a 1-in. discharge line, *G* is connected into the suction line of the feed and contains a 1-in. swing check valve.

C. N. WILEY.

Pittsburgh, Penn.

Open-Circuited Compensator

The diagram represents the connections of a three-phase compensator for starting squirrel-cage induction motors. The leads from the line and to the motor are flexibly connected to the rocker-drum contacts marked "off." When the handle is in the position marked "start," the rocker makes the connections indicated by the double dotted lines; and with the handle in the running position, the connections are as indicated by the double full lines. In the starting position the line wires connect to the autotransformer terminals and the motor leads to the autotransformer taps, thereby applying approximately half voltage to the motor. In the running position the line wires are connected directly to the motor leads so



COMPENSATOR CONNECTIONS

that the motor gets full voltage. The autotransformer is then entirely cut out.

It will be noted that the fuses are in circuit only on the running side. Inspection of the diagram also shows that the three wires that connect the cylinder contacts to the motor are in use irrespective of the position in which the starter may be; therefore, an open-circuit that affects operation in both positions is either local to these motor wires or to the line wires beyond the compensator connections.

An operator complained that his motor would not start from either side of the compensator, but would buzz, thereby indicating single-phase operation. Just above the compensator there was a small panel carrying the fuses and studs by means of which the compensator wires were connected to the line wires and to the fuses and motor wires. Connections were made with terminals similar to that indicated in Fig. 2, the ends of the wires being soldered into the sleeves and the eyes being slipped

onto the studs and held there with nuts. With a bell the open circuit was finally located in the sleeve of the terminal wire marked *X*, Fig. 1. This had been caused by a layer of resin between the wire and the sleeve, the resin having been used as a soldering flux.

J. A. HORTON.

Schenectady, N. Y.

Notes on Indicator Diagrams

In the Mar. 2 issue, A. R. Nottingham, under the above caption, discusses freak diagrams, particularly one that appeared in POWER for Nov. 3, 1914. He states: "In the original diagram Fig. 1, the atmospheric line is 5½ lb. too high. It should be where the dotted line is, though this does not affect the diagram so far as valve analysis is concerned."

Granting that the diagram was taken from the low-pressure cylinder, since neither the scale of the spring nor the vacuum line is given, how did Mr. Nottingham arrive at the conclusion that the atmospheric line is 5½ lb. too high? As a matter of fact the diagram was claimed to have been taken from the high-pressure cylinder of a compound engine, and the atmospheric line should have been below the exhaust line, equal to the receiver pressure, which could not have been high in this case as most of the work was done by the high-pressure cylinder.

VICTOR BONN.

New York City.

Boiler of a Traction Engine Exploded

The photographs reproduced show the general appearance of a traction engine after the boiler had exploded. The age of the boiler and the condition of the safety valve and steam gage are unknown to the writer. The shell was badly corroded and in one place had leaked and been patched, as may be seen at the lower part of Fig. 3. The weakest point seems to have been at the edge of the seam which joined the body to the firebox, where the sheet was corroded nearly through. There was but little scale on the surfaces.

The explosion occurred while the engine was standing on the public highway. The shell was carried 300 yards by the force of the explosion and the whistle was found one-quarter of a mile away. Fortunately, no one was hurt.

Claflin, Kan. J. J. BEEMAN.

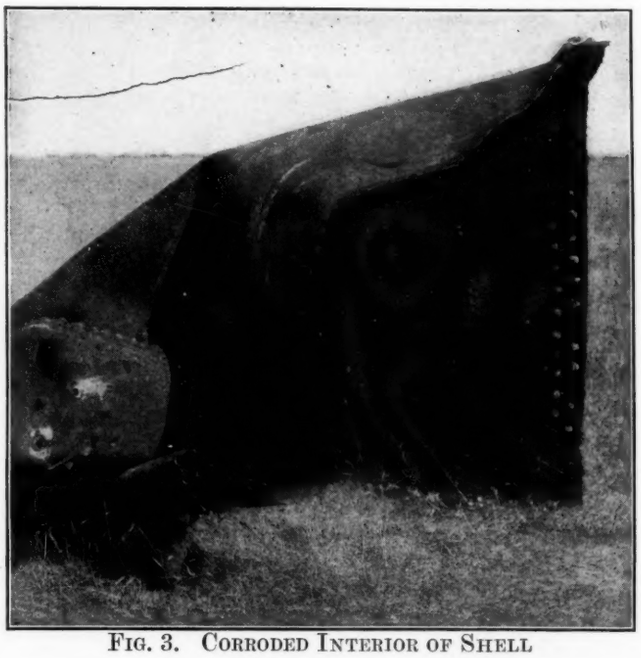


FIG. 3. CORRODED INTERIOR OF SHELL

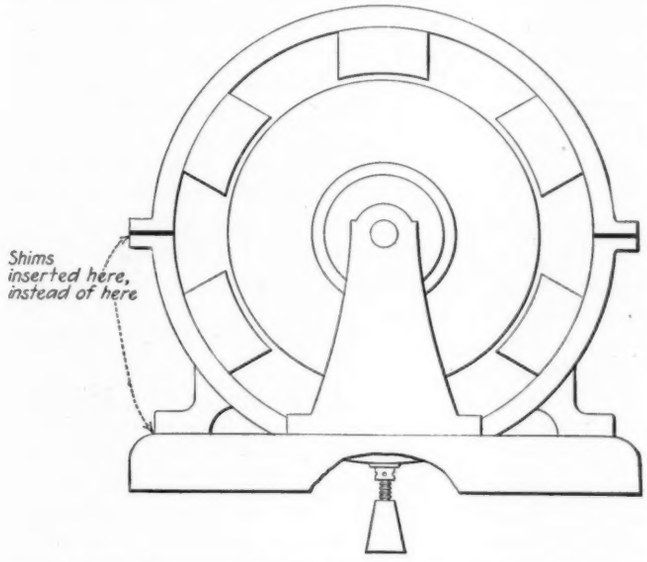
FIG. 1. PART OF THE WRECKAGE

FIG. 2. SMOKEBOX END

Unequal Air Gap

Some time ago the writer had some experience with a generator failing to operate correctly, the cause of which may interest readers of POWER.

The machine was a 100-kw. 220-volt shunt-wound generator direct-connected to a gas engine, and operated various amusement devices and lights at a summer resort. The rated voltage would build up readily, but as soon as any load was connected, the voltage would fluctuate and the brushes spark badly, and no amount of shifting would remedy the trouble. The engine was checked for its



FRAME SHOWING WHERE SHIMS WERE INSERTED

rated speed as shown on the nameplate and found correct, and no appreciable variation was noted when load was applied.

Upon inquiry it was found that the generator had just been installed, having been purchased second-hand. The field frame was split horizontally, and in order to facilitate handling and shipment to its present location the generator had been taken apart. As the men who had set it up were not experienced in handling electrical machinery, the natural conclusion was that possibly the field coils had been wrongly replaced or the brushes improperly spaced.

The fields were tested for polarity and found correct, and the spacing of the brushes was also gone over. It was while the armature winding was being examined to determine the neutral position of the brushes, that the unusual size of the air gap became apparent. Further examination of the frame revealed the presence of shims between the halves of the frame. The men who had done the erecting then explained that after they had placed the top half of the frame in position they noticed that there was just sufficient room beween the top of the armature and the bottom of the field coils to prevent rubbing while the armature was turned, whereas the space between the lower side of the armature and the bottom coils was quite liberal. So they had cut up some sheet iron and inserted it between the halves of the frame to make the air gap on the top of the armature equal that at the bottom. Here was the cause of the trouble.

A hoist was rigged from one of the roof trusses above the generator, the bolts holding the halves of the frame were taken out and the upper half was raised enough to remove the offending shims. The halves were then tightly bolted together and the air gap at the top and bottom carefully gaged.

The bolts holding the frame to the base were then loosened, and while the frame was steadied on all sides a jack, placed as shown in the sketch, raised it enough to insert sufficient shims to make the air gap equal at top and bottom. This completed, the engine was started again and the generator carried full load without further trouble.

The next morning one of the men found the shims that were sent with the machine and that should have been placed between the base and frame in the first place, securely tacked inside of the crates, out on the rubbish heap.

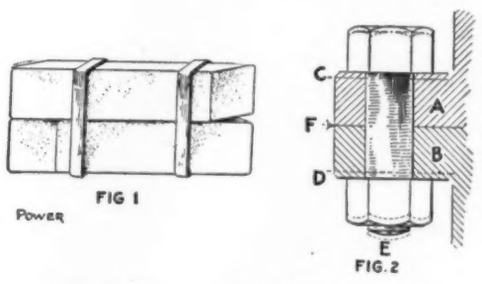
P. JUSTUS.

Cleveland, Ohio.

Holding Power of a Bolt

In deciding the size of a bolt to be used it should be assumed that it may be tightened by a helper who is instructed "to get it tight." To him this means that he is to put his whole weight on the wrench, if not extend the handle with a piece of pipe as long again.

To get a simple analogy to the action of a bolt clamping two pieces of metal together, take two erasers and fasten them together with a pair of light elastic bands, Fig. 1. Just as soon as a pull is exerted on the erasers they come apart enough to show light between them. The bands are more elastic, so they yielded more than the erasers were compressed by them in the first place. If very soft erasers and heavy bands are used, then quite a pull can be exerted on the erasers before any light will show, because they will go back to their original thickness before the same pull has stretched the elastics that much. This explains why it is necessary to have joint packings with some give to them to hold pressures. If the flanges and the bolts were of the same material it would be theoretically impossible to



RUBBER BAND AND BOLT ANALOGY

make up a tight joint without packing. Practically, it is feasible because the pressure of the bolts is hardest close to the bolt itself and the metal around the bolt hole is compressed more than that further away and more than the bolt is stretched.

Fig. 2 shows this more clearly. The full lines show a bolt and nut drawn up just tight enough to touch the two flanges *A* and *B*. When the nut is drawn down hard the flanges are each compressed to the dotted lines *C* and *D* (exaggerated, of course) and the bolt itself is stretched to the line *E* at the same time. No matter how tightly the bolt is set up, it will continue to lengthen with addi-

tional pull. Now suppose that some pressure is brought to bear to separate the two flanges. It will stretch the bolt, but every particle that the bolt stretches, the flanges, which were compressed, tend to go back to their original thickness, so that there is a tight joint until the flanges have got back where they started from; then the opening is due to the additional stretch of the bolt. If an elastic packing is inserted at *F* it will also keep the joint tight until all the pressure on it due to the bolts is released. A corrugated copper gasket acts in the same way, the corrugations, which are flattened out under pressure, coming back when the pressure is relieved.

The longer the bolt, the more it will stretch under a given pull. An eight-inch bolt will stretch four times as much as a two-inch, but the packing has only so much give in it, hence the decided advantage in using short bolts where possible. The stretch also decreases as the square of the diameter increases. A one-inch bolt will hold four times as rigidly as a half-inch; not merely four times as much in pounds, but with only one-fourth the stretch for a given load.

There is, however, an advantage in a comparatively long bolt in places where it can act as a relief. If a cylinder is full of water the bolts may stretch enough to let it out without further injury, but, aside from this emergency, the use of short, thick bolts is good practice.

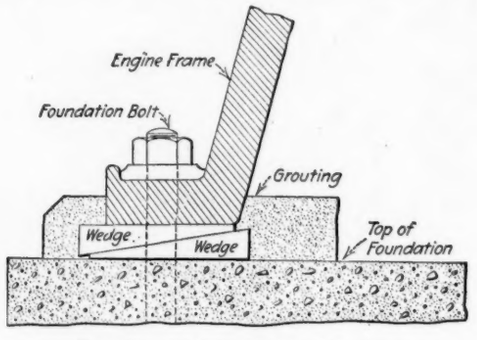
E. H. Fish.

Worcester, Mass.

Grouting under Heavy Machinery

To the articles under this title in the issue of Mar. 2, page 310, and Apr. 6, page 482, much can be added and still leave the matter of grouting unsettled in the minds of many.

My belief is that at least an inch should be left between the top of the foundation and the machine, and that the



Another System of Wedges

latter should be leveled by means of wedges placed as shown on the illustration, one set on each side of every foundation bolt. When the leveling is complete, every bolt should be tightened, so as to hold the machine tightly against the wedges. The latter should all be left permanently under the machine; and they will not slip if they are made of cast iron with the surface left rough.

The grouting material may be either neat cement, a mixture of sand and cement, sulphur, or lead. It should be poured so that it will cover the top of the foundation under the machine, and also cover the ends of the wedges on the outside, as shown. If a mixture of sand and cement is used it should be sufficiently thin to flow easily. On account of the cost, lead is seldom used, but it makes a very satisfactory job. Sulphur cannot be used under machines where the heat will be so great as to melt it.

J. E. Poche.

New Orleans, La.

The Boiler Inspector Confesses

I have read the lurid description of a boiler inspector's confession as printed on the Foreword page of the Mar. 9 Power. There is one item in it which causes me to wonder how it ever "got by" the editorial force.

It is this: "Investigation showed that the fireman had opened the blowoff valve, and before closing it rushed away on a signal from the engineer. Before he returned the explosion had happened. This knowledge relieved my mind, but I had learned my lesson."

It is needless to tell you that an explosion might possibly be caused by a sudden opening of a blowoff valve, but if the valve had been open for any length of time and left open, as in this case, the pressure would have been gradually lowered and the boiler would not have exploded.

Charles H. Garlick.

Pittsburgh, Penn.

[The chance of avoiding the rupture of a boiler depends on the pressure being reduced to nearly zero before the sheet or sheets become bursting hot. This may happen with the blowoff valve open. But if two or more boilers under pressure are connected to a common header and one bursts a sheet because of low water, a violent explosion would likely follow, blowoff open or not. There was more than one boiler in the plant mentioned in the foreword.—Editor.]

Pulsations in Steam Pipes

Correspondence published in the columns of Power has quite frequently referred to the methods which have been employed, with varying success, to cure pulsations of steam pipes. The remedy most commonly suggested is to place a receiver of moderate size in the steam pipe near the engine.

I have designed and had built several receivers of varying capacities, the smallest having a volume twice that of the cylinder which it supplied. Others were much larger, but not one was big enough to show any appreciable diminution in the vibrations in the line. However, there were advantages gained, which more than paid for the work and expense of putting them in, such as drier steam, higher initial pressure and a better steam distribution in the engine cylinder.

But in no instance did I succeed in stopping the vibrations entirely, without throttling the supply to the receiver until the flow in the steam pipe was comparatively steady.

I have seen just one receiver placed in the steam line to an engine that was large enough to stop the vibrations entirely, and that one, I am sorry to say, I did not and would not at that time dare to recommend.

Two boilers cross-connected by a drum riveted to the shells supplied steam to a 14x42-inch Corliss engine running 80 r.p.m. The four-inch steam pipe to the engine was about forty feet long, and vibrated in spite of anchors

to such an extent that calking and reriveting the drum nozzles was a serious matter.

The solution of the problem was finally left to the engineer, who had the steam drum (which was 42 in. dia. by 14 ft. long) taken from the boilers, set vertically and connected direct to, and placed directly at the top of, the throttle valve of the engine. There was no further trouble from vibration or leaking rivets, and the initial pressure in the cylinder was six pounds higher than before the change was made.

That was the only receiver I ever saw large enough for the purpose intended, and the only one I know of that proved to be a complete solution of the pulsation nuisance.

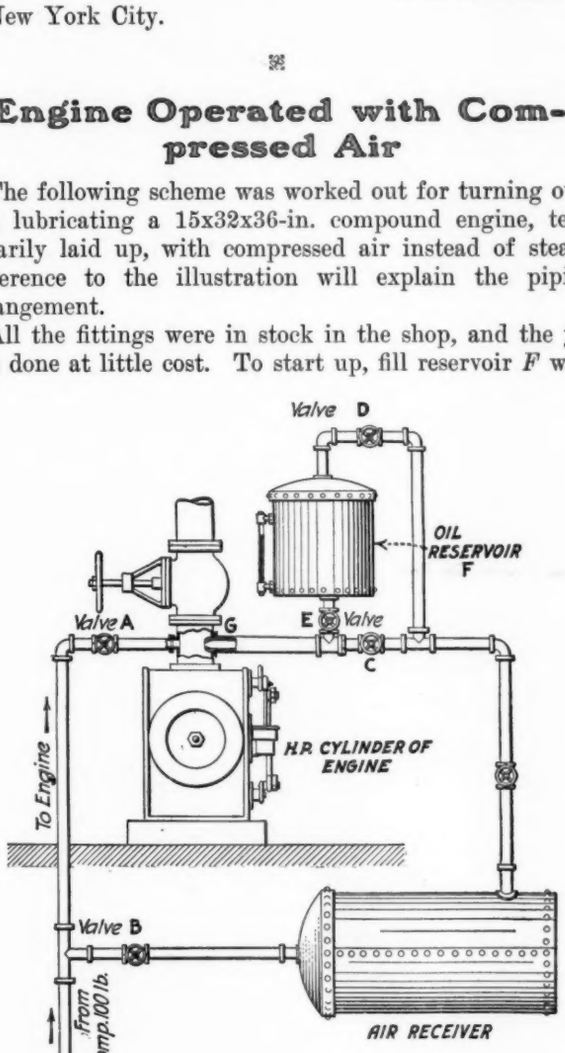
F. L. JOHNSON.

New York City.

Engine Operated with Compressed Air

The following scheme was worked out for turning over and lubricating a 15x32x36-in. compound engine, temporarily laid up, with compressed air instead of steam. Reference to the illustration will explain the piping arrangement.

All the fittings were in stock in the shop, and the job was done at little cost. To start up, fill reservoir *F* with



PIPING ARRANGEMENT FOR COMPRESSED AIR

light cylinder oil, open starting valve *A*, close receiver valve *B*, then open valves *C* and *D* slightly. Regulate the oil with valve *E*.

After the engine starts there is a fall of pressure in the line from the compressor on account of wire drawing in the small line. The receiver being closed off, the pressure does not drop rapidly and is higher at *C* than at *A*, thereby causing the atomized oil to flow through nozzle *G* into the engine.

C. H. REED.

East Chicago, Ind.

Telephone Receiver Connected to Calipers

A telephone receiver has been my constant friend about the plant for the past twelve years, for various uses, one of which is in connection with calipering, especially the work inside of engine cylinders, as in Fig. 1. The most convenient way is to have the two sides of the calipers insulated from each other. In Fig. 2, where the work in the lathe completes the circuit, causing a click in the receiver, an ordinary pair may be used with a cigarette paper between the work and one side of the calipers. This method is especially useful in aligning engines.

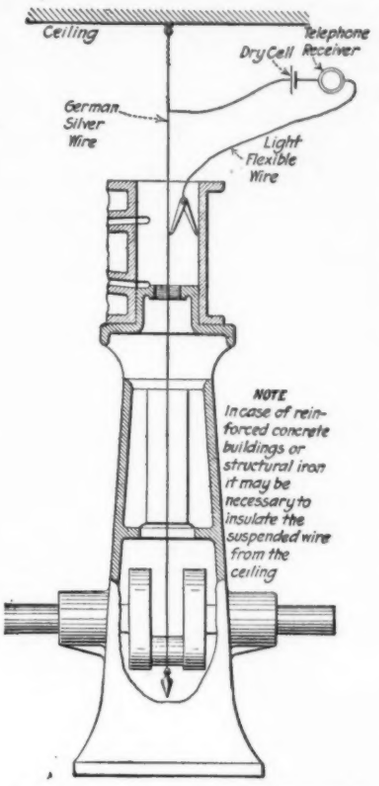

Ceiling
Telephone Receiver
Dry Cell
German Silver Wire
Light Flexible Wire
NOTE
In case of reinforced concrete buildings or structural iron it may be necessary to insulate the suspended wire from the ceiling

FIG. 1. ALIGNING AN ENGINE

Use a slender german-silver wire for the line, taking care that it is insulated from the ground if an ordinary caliper is used, or put a cigarette paper next to the cylinder wall; then, with one side of the circuit connected to the aligning wire and the other to the calipers, a circuit will be completed when the two come in

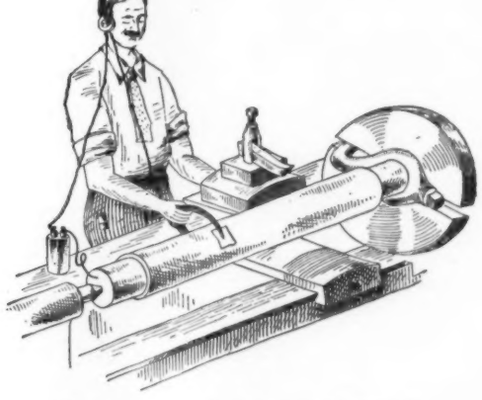


FIG. 2. GAGING OUTSIDE WORK

contact. A click will be heard when a contact is so slight that it cannot be seen or felt. Care should be taken not to use too much battery power, as it is annoying to the ear.

AMOS J. CARR

Fort McKinley, Me.

Heating Surface of Superheaters—Experiments prove that 5 B.t.u. per sq.ft. of heating surface will be transferred to steam for each degree of difference between it and the gases.

Easy Calculation of Steam Coal Required by Power Plants

By R. E. Horton*

The writer and his assistants have frequent need to calculate the cost of coal required by actual or hypothetical steam plants under comparison with proposed hydraulic stations. After a few laborious repetitions of computations going back to fundamental factors, the office practice was standardized in the interest of general economy and capacity. A table prepared is here given as of possible interest to others.

Computations were carried through and tabulated for the yearly coal consumption in tons at a rate of 1 lb. per hp.-hr. under various conditions. Now it is only necessary to ascertain or estimate and combine (1) the simplest unit coal consumption (per horsepower-hour, including allowance for shrinkage and waste if any); (2) the average horsepower in use when running; (3) the allowance for banking; (4) the hours' use per day, and days per year.

FACTORS FOR CALCULATING AMOUNT OF STEAM COAL REQUIRED PER HORSEPOWER-YEAR

	Gross Tons, 2240 Lb.		Net Tons, 2000 Lb.	
Method of Operation	310 Days	365 Days	310 Days	365 Days
10 hr. per day, no banking.....	1.38	1.63	1.55	1.83
10 hr. per day plus ¼ for banking	1.84	2.17	2.07	2.43
12 hr. per day, no banking.....	1.65	1.96	1.86	2.19
12 hr. per day plus ¼ for banking	2.21	2.61	2.48	2.92
24 hr. per day, no banking.....	3.32	3.91	3.72	4.38

For example: A plant runs 10 hr. per day and 310 days per year, produces 500 hp. average, uses 2½ lb. per hp.-hr. of steam coal, has ¼ allowance for banking; coal costs $3.50 per gross ton. From the table, the proper unit consumption per horsepower year is 1.84 gross tons.
Then,

$$2.5 \times 1.84 \times 500 \times \$3.50 = \$7735 \text{ annual cost.}$$

Sometimes it is necessary to know the tons of ash that will have to be disposed of each year; then it is necessary only to substitute the decimal percentage of ash in the coal for the price per ton. For 15% ash the foregoing case shows

$$2.5 \times 1.84 \times 500 \times 0.15 = 345 \text{ gross tons.}$$

—"Engineering News."

Aneroid Calorimeter†

By H. C. Dickinson and N. S. Osborne

The term aneroid calorimeter is applied to a type of instrument in which equalization of temperature is secured by means of the thermal conductivity of copper instead of by the convection of a stirred liquid. The calorimeter, consisting of a thick-walled copper cylindrical vessel in the walls of which are embedded a coil of resistance wire to supply heat electrically and a platinum resistance coil for use as a thermometer, has been found useful over a wide range of temperatures and is applicable to a variety of problems.

For use at low temperatures the device is mounted in a jacket surrounded by a bath of gasoline, the temperature of which can be controlled thermostatically to within a few thousandths of a degree at any temperature between —55 and +40° C. or can be changed rapidly in order to keep it the same as that of the calorimeter when heat is being supplied to the latter.

Differences in temperature between the surface of the calorimeter and that of the jacket are measured by means of multiple thermocouples which have 10 junctions distributed over the surface of each, thus making it possible to apply accurate corrections for thermal leakage between the calorimeter and the jacket even when the temperatures of both are changing rapidly.

The platinum resistance coil (for use as a thermometer) embedded in the calorimeter shows slight irregularities in its behavior, probably due to the difference in expansion between the platinum and the copper which surrounds it. Uncertainties on this account, while in general negligible, can be avoided by measuring the temperature of the outer bath with a standard resistance thermometer, using the thermocouples to measure the small difference, usually not more than a few thousandths of a degree, between the calorimeter and the jacket. The thermometer could probably be improved by changing the construction.

Results of a series of experiments (page 565, Apr. 27, 1915, issue) give the constants of the resistance thermometer and the heat capacity of the calorimeter, including a tin-lined cell for use in determining the specific heat of ice and water and the latent heat of fusion of ice.

A series of check experiments on the specific heat of water shows the order of reproducibility of results which can be obtained with this calorimeter to be 1 part in 2000. Measurements made at temperatures between 0 and 40° C. gave results which agree to within the limits of experimental accuracy with the unpublished results of a long series of experiments made in the usual form of stirred-water calorimeter. The results are also in satisfactory agreement with the most probable values deducible from the data of the most careful investigations published by other observers.

*Consulting Hydraulic Engineer, 57 No. Pine Ave., Albany, N. Y.

†Abstract of a paper issued by the Bureau of Standards, Department of Commerce, Washington, D. C.

Influence of Indicator Connecting Pipes*

By Thomas W. Morley

It is usually stated that one of the conditions for accuracy in taking engine-indicator diagrams is that the connection between the engine and the indicator should be short, direct and of ample bore. The effect of bends, undue restrictions and excessive length of indicator pipes would be to delay the pressure change at the indicator, and hence to cause errors in the diagram. This resistance can best be investigated experimentally, and the author, having been able to find only very meager evidence as to the errors introduced by various connections, has conducted experiments to find out whether the arrangements commonly used are responsible for measurable errors.

A small steam engine was used to produce cycles of pressure change typical of those occurring in steam-engine practice. The engine employed was vertical, with a 6½-in. cylinder, 6-in. stroke, and ordinary slide-valve and link-motion valve gear. The cutoff was kept at about 0.4 of the stroke throughout the experiments.

Alternative indicator connections, long and short (Fig. 1), were arranged, and pairs of diagrams taken through these connections were compared. A and B denote the points at

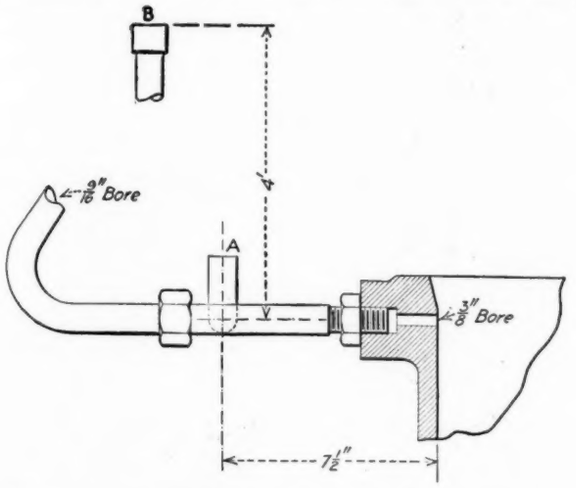


Fig. 1. Alternative Indicator Connections

which the indicator cocks were coupled. The upper end of the long pipe was, of course, suitably fixed.

At first, two similar indicators were placed at A and B, but it was found that, although of the same make, they did not give diagrams that admitted of convenient comparison. In subsequent experiments only one was used, care being taken that no change in the steam pressure or engine speed took place while the indicator was transferred from one position to the other.

Before the arrangement shown in Fig. 1 was adopted, preliminary experiments were made, using next the engine a two-way cock connection. The length between the upper and the lower positions was the same as in Fig. 1.

*From a paper presented before the Institution of Engineers and Shipbuilders in Scotland.

The diagrams thus obtained differed widely. For example, at 325 r.p.m. and a maximum pressure of 50 lb. per sq.in., the mean effective pressures were 36.4 for the lower and 20.4 lb. per sq.in. for the upper diagram. Even at 80 r.p.m. the difference was about 30 per cent. This was caused chiefly by the small bore ($\frac{1}{16}$ in.) of the two-way cock. After it was bored out to $\frac{3}{8}$ in. the difference of the diagrams fell to 4 per cent. at about 300 r.p.m. The use of the two-way cock still had the drawback that, when the upper indicator was in use, the pipe connection formed an appreciable addition to the cylinder clearance volume. This in itself might affect the cycle of pressure change in the engine and so account for the difference in the diagrams.

In the final arrangement, shown in Fig. 1, the pipe connections to each indicator cock were constantly open to the cylinder, so that the pressures in the whole system were al-

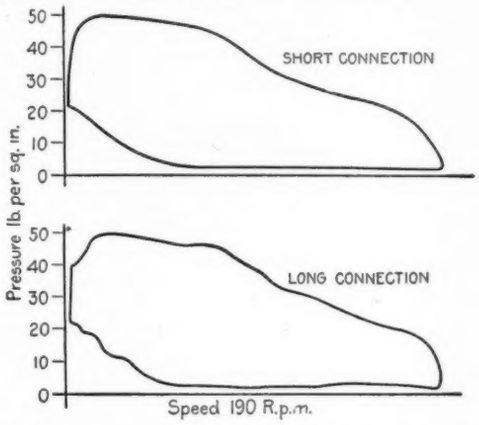


FIG. 2. INDICATOR DIAGRAMS FROM ALTERNATIVE CONNECTIONS

ways unaffected. Also, except for the short $\frac{3}{8}$-in. channel at the cylinder wall, the connections to the indicators were of uniform bore. Under these conditions a number of tests were made with gage pressure up to 75 lb. per sq.in., and speeds up to 280 r.p.m. Fig. 2 shows a typical pair of diagrams. The differences between them are so small that they would be concealed if one had been superposed on the other. The chief visible difference in the diagrams was that those taken with the long connection showed more waviness of outline, due, no doubt, to the oscillations set up in the long pipe.

The following table shows the results of the experiments:

Speed of Engine R.p.m.	Maximum Pressure (from Diagram) Lb. per Sq.In.	Mean Effective Pressure Long Connection P_1 Lb. per Sq.In.	Mean Effective Pressure Short Connection P_2 Lb. per Sq.In.	Difference $P_1—P_2$ Lb. per Sq.In.	Difference per Cent.
150	48	27.32	27.48	—0.16	—0.6
153	29	15.46	15.46	...	...
178	76	49.44	49.44	...	...
182	73	49.9	49.9	...	...
190	25	13.15	13.15	...	...
190	31	17.75	17.6	+ 0.15	+ 0.85
195	40	24.4	24.0	+ 0.4	+ 1.6
190	48	29.65	29.9	—0.25	—0.85
195	55	35.1	35.1	...	...
190	60	38.2	38.2	...	...
190	60	38.6	38.2	+ 0.4	+ 1.1
190	68	44.9	44.9	...	...
272	25.5	12.24	12.44	—0.2	—1.6
240	70	43.2	42.8	+ 0.4	+ 0.95
254	66	42.12	43.02	—0.9	—2.1
284	38.5	22.1	21.5	+ 0.6	+ 2.8
280	42	24.8	25.2	—0.4	—1.6
275	47	27.7	28.1	—0.4	—1.4
280	53	32.6	32.3	+ 0.3	+ 0.95
270	62	37.9	37.8	+ 0.1	+ 0.25

The general conclusion to be drawn from the experiments is that the influence of the connecting pipes does not subject indicator diagrams to any appreciable error, except in abnormal cases. The difference of the mean effective pressures was always small, in only two cases exceeding 2 per cent.; and was in fact within the limit of probable error of the indicator itself.

In the discussion following the delivery of the paper the conclusions reached were questioned because the connection at A in Fig 1 had two bends and the long pipe was in communication with the cylinder while the indicator at A was in use. In answer to the latter objection the author explained that the arrangement used satisfied the real object of the experiments, which was to compare the influence of long and short pipes connected to a point at which a certain pressure cycle took place. The indicator at A need not give a faithful record of the cylinder pressures. Coming now to the other objection, the indicators at A and B were separated from the point where there was a constant cycle of pressures by a bend and a short pipe in one case, and in the other by an easier bend and a long pipe. Experiments other than those described in the paper had been made on this engine with connections of about the same length as that to A, but with a different number of bends. No appreciable difference in the results had been observed. It was therefore considered that any difference in the influence of the two connections would be due to their lengths alone and would not be obscured by the bends.

Why Should Such Things Be?

A leading machinery firm in Bombay expressed the opinion that American boilers could not be successful in India, because of the steel used not being of the right quality as required by government testers. American manufacturers do not follow Indian specifications, and offer openhearth steel instead of basic. The market in India is for boilers built for high pressure. The steel should not be too hard, but somewhat flexible. German boilers have sold well in India because they are made of steel that expands well. Nearly all boilers used in India are of Lancashire type. This firm states that it imported a boiler from the United States about 15 years ago and still has it on hand, being unable to sell it. Boilers in India are mostly used for cotton mills, ginning factories, etc. —Henry D. Baker, U. S. Consul, Bombay.

Blowout of Turbine Exhaust Casing

Although the local newspapers reported that an accident had taken place on Mar. 31 at the plant of the Boston Manufacturing Co., Waltham, Mass., the causes given varied from a blowout of the cylinder head of a turbine engine to a turbine explosion. Perhaps the most comprehensive was the statement that the explosion in a 750-kw. engine blew up the generator and badly damaged the boiler.

Further investigation has shown that at 5:45 p.m. on the date mentioned, the exhaust casing on a 750-kw. G-E Curtis turbine was blown to pieces, hurling through a window the engineer, William Finley, who was standing near. Outside of the turbine the damage was confined to the wrecking of the engine-room windows and the dislodging of a few bricks by pieces of the casing blown against them.

The accident is said to have been caused by the closing of the exhaust valve before the throttle was closed, the relief valve provided for such emergencies failing to operate.

PERSONALS

Messrs. John S. Griggs, Jr., and David Moffat Meyers have consolidated their consulting practices with offices at 110 West Fortieth St., New York City. Mr. Griggs was a member of the consulting firm of Griggs & Holbrook, New York, and has been in practice for the past twenty years. Mr. Meyers, who is the author of the recently published book, "Preventing Losses in Factory Power Plants," was formerly mechanical engineer for the United States Leather Co. The firm will specialize in mechanical and electrical propositions for industrial and other installations.

C. P. Poole, chief engineer of the Department of Mechanical Engineering of the City of Atlanta, Ga., has been appointed a member of the International Jury of Award in the Department of Machinery of the Panama International Exposition. Mr. Poole, who will be remembered as one of the editors of "Power," has been for some time in charge of what was known as the Department of Smoke and Gas Inspection of Atlanta, and under his direction the activities of the department have broadened out so as to make necessary the reorganization indicated by the new title.

ENGINEERING AFFAIRS

Stationary Engineers' Day at the Fair—The management of the Panama-Pacific Exposition has designated Saturday, May 29, as Stationary Engineers' Day. This will enable the delegates and guests of the California State Association of

the N. A. S. E. to enjoy that day at the Fair, since the annual convention of this association will be held in San Francisco on May 27 and 28.

The Fifteenth Annual Summer Session of the College of Engineering of the University of Wisconsin will be held at Madison during the six weeks' period beginning June 21. Special courses will be given in electrical, steam and hydraulic engineering, gas engines, machine design, mechanical drawing, mechanics, shopwork and surveying. Further information may be obtained from F. E. Turneaure, Dean, Madison, Wis.

The International Engineering Congress which is to be held in San Francisco in September will issue in Vol. IV of its transactions the papers presented on "Railways and Railway Engineering." The field treated will cover the relation of railways to social development, present status of railways, economic factors governing building of new lines, location, physical characteristics of road including track and roadbed, bridges, tunnels, terminals, construction methods, signals, road equipment, including motive power other than electric, rolling stock, floating equipment, and electric motive power. W. A. Cattell, 417 Foxcroft Bldg., San Francisco, secretary of the Congress, will furnish particulars regarding membership and the securing of the transactions to those interested. These transactions will include nine or ten volumes, covering the various fields of engineering.

NEW PUBLICATIONS

MECHANICAL WORLD POCKETBOOK, twenty-eighth issue. Norman Remington Co., Baltimore, Md., American agents. Size, 4x6 inches; 330 pages. Price, 50 cents.

Besides the usual data given in previous editions of this well-known low-priced handbook, this issue contains an extended and rewritten section on toothed gearing and a more lengthy section dealing with structural iron and steel work. Also, new sections on limit gages and on the cost of power have been added.

ADVANCED ELECTRICITY AND MAGNETISM. By W. S. Franklin and Barry MacNutt. Published by the Macmillan Co., New York. Cloth, 5½x8½ inches; 300 pages; 217 illustrations. Price, $2.

This book is designed for students in colleges and technical schools. The subject matter deals with the elementary theory of magnetism and with advanced principles of the magnetic measurement of current, electromagnetic induction, electrostatics and electric waves. An outline is given of the electron theory and of its application to the vacuum tube and the electric arc. The historical development of the theory is entirely, and the mathematical largely, omitted, the treatment being confined to concrete presentations of a few principles. A number of problems (with answers) are given at the ends of the chapters.

HEATING AND VENTILATING BUILDINGS. By R. C. Carpenter. Published by John Wiley & Sons, Inc., New York. Cloth, 6x9 in.; 598 pages; 290 illustrations. $3.50.

Professor Carpenter's well-known treatise, now published in its sixth edition, is a manual for heating engineers and architects. It gives general methods of design and construction as well as the elementary principles applying to heating and ventilating apparatus. While the previous edition has been rearranged to a considerable extent, the additions in the sixth edition consist mainly of a chapter on air conditioning and of abstracts of heating and ventilating laws. The determination and regulation of humidity and the methods of purifying air are broadly described. Abstracts of the laws relating to the heating and ventilation of schoolhouses and public buildings in 17 states are given. The book has the disadvantages common to most technical treatises in which an attempt is made to keep them up to date by publishing new editions. Even with the best of intentions it is difficult to thoroughly revise an old book. This is evidenced in the present instance by the many results of tests and investigations made in the late 80's and early 90's. This material was adequate when the book was first published in 1895, but at the present time, although perhaps of historical value, is surely not indicative of the state of the industry. Another disadvantage is the difficulty in adding new material without affecting the unity of the original treatment. For example the present volume has two sets of sections numbered from 56 to 72 inclusive. Of course this is caused by carelessness in revision, but it goes to show a minor trouble experienced in forcing the original text to assume a new and different form.

TRADE CATALOGS

E. Keeler Co., Williamsport, Penn. Catalog. Return tubular boilers. Illustrated, 46 pp., 7½x10½ in.

Armstrong Cork Co., Pittsburgh, Penn. Pamphlet. "Good Furnaces Made Better." Illustrated, 20 pp., 3½x6 in.

The DuBois Machine Shop, 118 Hudson Ave., Albany, N. Y. Booklet. Randerson automatic piston ring. Illustrated, 3x6 in.

Otis Elevator Co., Eleventh Ave. and 26th St., New York. Catalog. Gravity spiral conveyors. Illustrated, 56 pp., 6x9 in.

B. F. Sturtevant Co., Hyde Park, Boston, Mass. Bulletin No. 214. Turbo-undergrate blower. Illustrated, 24 pp., 6½x9 in.

Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin No. 1532. Allis-Chalmers oil engines, Diesel type. Illustrated, 16 pp., 8x10½ in.

The D. T. Williams Valve Co., Cincinnati, Ohio. Catalog No. 10. Valves, steam cocks, water gages, lubricators, steam traps, etc. Illustrated, 320 pp., 5½x8 in.

American Blower Co., Detroit, Mich. Bulletin No. 24—Series 4. Sirocco heating, ventilating, cooling and purifying system. Illustrated, 32 pp., 8½x11 in.

Gas Engine & Power Co. and Chas. L. Seabury & Co., Cons., Morris Heights, N. Y. Catalog No. 10. Seabury safety water tube boilers. Illustrated, 46 pp., 6x9 in.

Richardson-Phenix Co., Milwaukee, Wis. Bulletin No. 10. Peterson power plant oil filter and accessory apparatus for central oiling systems. Illustrated, 32 pp., 8½x11 in.

General Electric Co., Schenectady, N. Y. Bulletin No. 48,904. Electric arc welding. Illustrated, 10 pp., 8x10½ in. Bulletin No. 48,905. Arc welding apparatus. Illustrated, 6 pp., 8x10½ in.

Spray Engineering Co., 93 Federal St., Boston, Mass. Bulletin No. 101. Sprays for Cooling Condensing Water. Illustrated, 14 pp., 6x9 in. Bulletin No. 151. Washing and Cooling Air for Steam Turbine Generators. Illustrated, 8 pp., 6x9 in.

Classified Ads

Positions Wanted, 3 cents a word, minimum charge 50c. an insertion, in advance **Positions Open,** (Civil Service Examinations), **Employment Agencies** (Labor Bureaus), **Business Opportunities, Wanted** (Agents and Salesmen—Contract Work), **Miscellaneous** (Educational—Books), **For Sale,** 5 cents a word, minimum charge, $1.00 an insertion.

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Original letters of recommendation or other papers of value should not be inclosed to unknown correspondents. Send copies.

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MASTER MECHANIC for rolling mill. P. 487, Power.

A CENTRIFUGAL PUMP DESIGNER with experience in designing high-speed pumps of small and medium sizes for high- and low-head service; applicants must state fully their experience, age and salary expected. P. 491, Power.

POSITIONS WANTED

CHIEF ENGINEER, employed in central station; seven years' experience with engines, turbines, dynamos, boilers; married; age 30. P. W. 488, Power, Chicago.

ENGINEER, competent to take full charge of industrial power plant; familiar with usual types of steam and electric equipment, refrigerating machinery, elevators, etc.; first-class references; available at once. P. W. 493, Power.

MAN of wide experience in power-plant, light-plant and water-works management, construction, installation and operation desires position; Middle West or West preferred; combustion engineer and all-round efficiency man; prefer to act as chief engineer or superintendent of power for company controlling several plants who wants better results; or would consider manager's position; at present employed at $2500 per year; nine years with present company; 34 years old; change necessary on account of wife's health; anything reasonable considered; no intoxicants, narcotics or profanity; references given and required. P. W. 484, Power, Chicago.

WANTED

AGENTS AND SALESMEN

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. W. 120, Power, Chicago.

SALESMAN selling engineers' supplies wants connection with reliable concern; Rhode Island, southern Massachusetts; salary and commission. W. 495, Power.